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"GUESS I'LL STICK AROUND A WHILE."

Power Plant of the J. B. Stetson Company

By Warren O. Rogers

SYNOPSIS—A plant in which more capacity was required. The problem was solved by putting in a mixed-pressure turbine, utilizing the exhaust steam from the engines and numerous pumps. The condensing water goes to a cooling tower which, owing to the restricted ground area, is supported on concrete posts in the yard.

When an engineer puts on his "Stetson," he gives but little thought to the power plant which made the manufacture of this hat possible or to the immense factory in which it was made. While the power plant of the J. B. Stetson Co., Philadelphia, Penn., is not new, it has interesting features, and illustrates how more power and greater economy may be obtained in a plant where the engine room cannot well accommodate additional units.

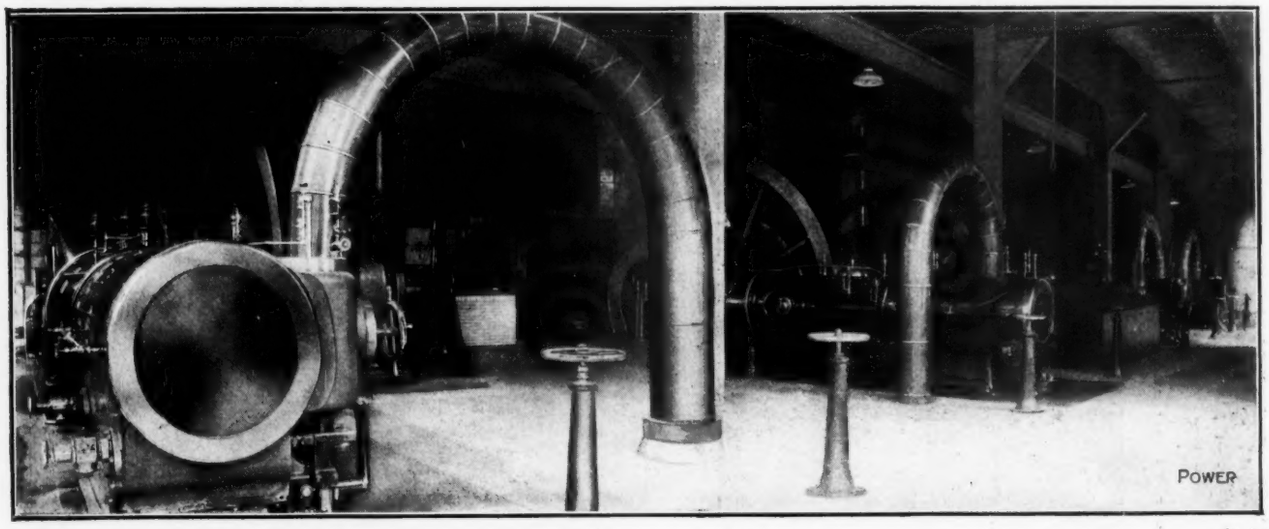


Fig. 1. General View of the Engine Room

Fig. 1 is a general view of the engine room, which houses three 24x48-in. and two 30x48-in. horizontal engines, one 8½x8-in. single-acting, vertical reciprocating engine, and a Rateau-Smoot mixed-pressure steam turbine which uses the exhaust steam from such engines as are run. Before the turbine was put in, both of the large units and two of the small ones were used, leaving one 250-kw. unit as a reserve. On the average these engines consumed 24½ lb. of steam per horsepower-hour. With the turbine, but one 500-kw. unit and one 250-kw. set are operated, with a large and two small units held as reserve. This means cutting out a 375-hp. and a 750-hp. unit, which at 24½ lb. of steam per hour represents a saving of 27,562 lb. of live steam per hour. Crediting the boilers with evaporating 9 lb. of water per pound of coal fired, a saving of 3062 lb. of coal would be had. This makes 1½ tons per hour, or 15 tons per day of 10 hours, and is a saving of $48.75 per day if the coal cost is $3.25 per ton.

As there are 14 boilers in the three boiler rooms, all connected to the same steam main, and as the turbine is a recent addition to the plant equipment, figures regarding the actual saving in fuel are not available. The foregoing, however, gives a fairly good idea of what the turbine is doing in the way of economy.

Exhaust-Steam Lines

The exhaust line from the two 30x48- and the two 24x48-in. engines begins with a 12-in. pipe and increases to 14-, 16-, 18- and 20-in., as indicated by Fig. 2. The other 30x48-in. engine exhausts into a 16-in. line, which also receives the exhaust from the steam pumps, and a 16x42-in. Corliss engine used to belt-drive a lineshaft in the pump room. This pipe line loops one end of the engine-room basement and joins the 20-in. main exhaust header. Exhaust steam is not only utilized by the turbine,

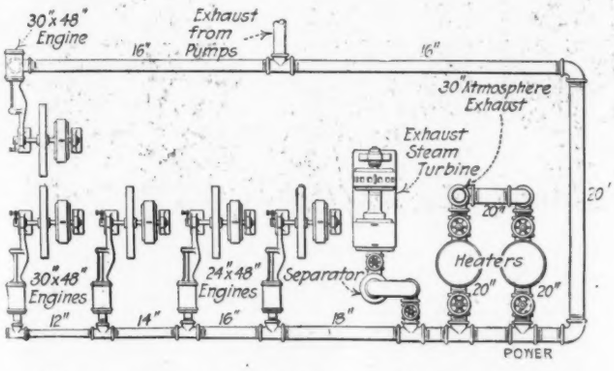


Fig. 2. Diagram of Exhaust-Steam Piping

but some is also used in the two 3000-hp. vertical heaters, which are piped as shown. Both are connected to a 30-in. atmospheric exhaust. The exhaust and the main steam lines in the basement (Fig. 3) are supported by brick

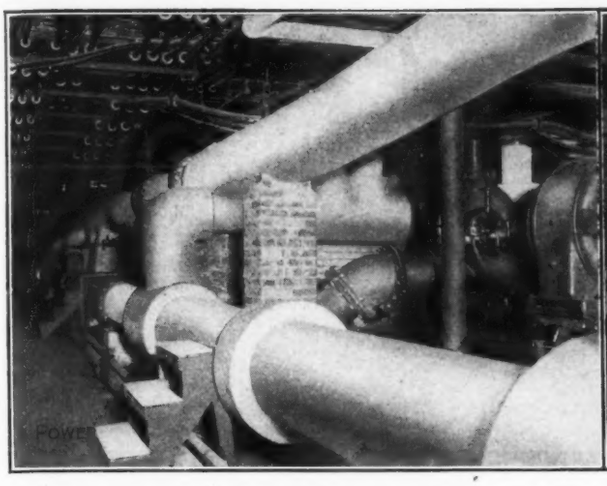


FIG. 3. LIVE- AND EXHAUST-STEAM MAINS

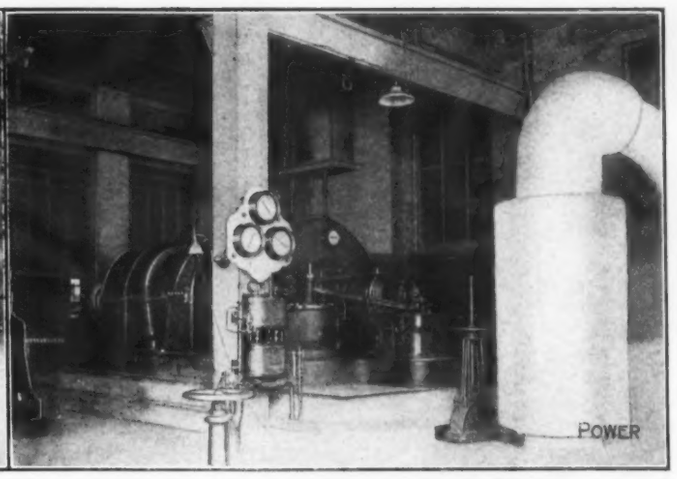


FIG. 4. LOW-PRESSURE STEAM TURBINE

piers; the lower pipe is the exhaust line. The motor-driven circulating pump supplies the surface condenser, which is further to the right, but not shown.

TURBINE AND CONDENSING APPARATUS

The low-pressure turbine (Fig. 4) is of 750-kw. capacity, generating 230-volt direct current at 1500 r.p.m. It is at the end of the engine room and rests on a concrete foundation. In the basement below the turbine is the condensing apparatus. The surface condenser has 4200 sq.ft. of cooling surface made up of 1-in. outside diameter No. 18-gage tubes. Condensing water is supplied by a 12-in. centrifugal pump, having a capacity of 3000 gal. per min., and being driven by a 70-hp., 220-volt, direct-current motor at 950 r.p.m. The "rotrex" air pump is driven by a 14-hp., 220-volt, direct-current motor.

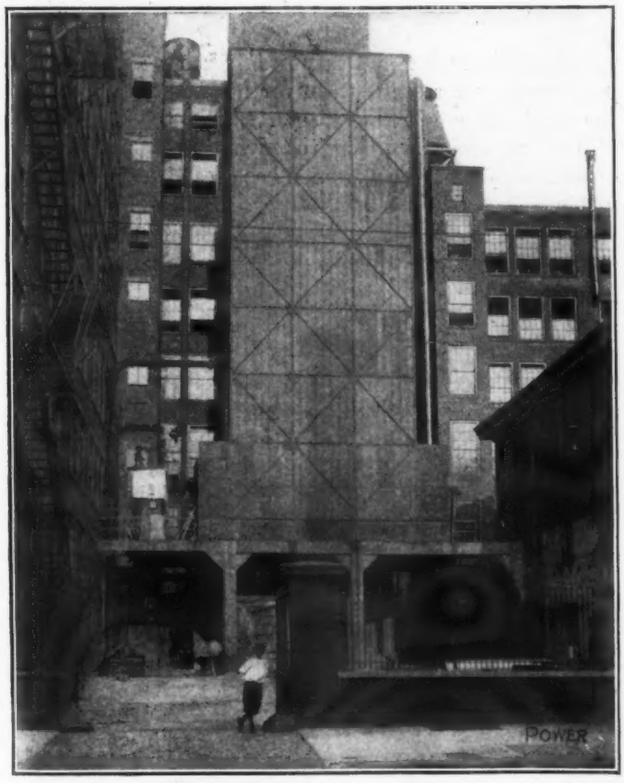


FIG. 5. COMBINED NATURAL- AND FORCED-DRAFT COOLING TOWER

Owing to a restricted ground area, the cooling tower is in the yard and rests on a concrete base supported by concrete posts (Fig. 5). The tower is 27x27 ft. and is 75 ft. high. The condensing water is cooled by a combination of forced and natural draft, the forced draft being supplied by four 8-ft. fans driven by direct-connected motors. The hot water from the condenser is used in the factory for manufacturing purposes, thus utilizing the heat imparted to it in condensing the steam, the temperature of the water being 90 deg. The water from the cooling tower goes to the condenser at 89 deg. during the ordinary summer temperatures.

There are more than 500 motors throughout the factory

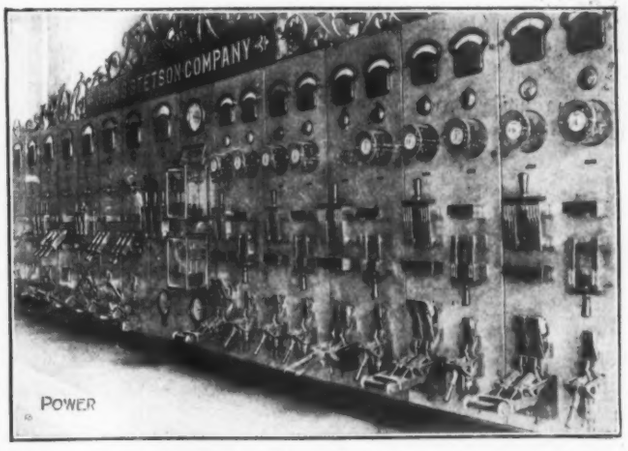


FIG. 6. DIRECT-CURRENT SWITCHBOARD

which range from $^1/_8$ to 70 hp. in capacity. The load on the turbine is from 270 to 3000 amp., and that on the engines is from 3000 to 3500 amp. There is an average load of 6000 amp., or about 1500 kw.

The motor and lighting circuits are controlled from a 14-panel marble switchboard (Fig. 6), the generator panels showing in the foreground. The turbine generator is controlled from a bench switchboard (Fig. 1), between the second and third engines.

BOILERS AND PUMPS

Steam is supplied for the engines and for manufacturing purposes by fourteen water-tube boilers in three rooms. Fig. 7 shows the five large boilers.

Coal is delivered by wagons from the street into two

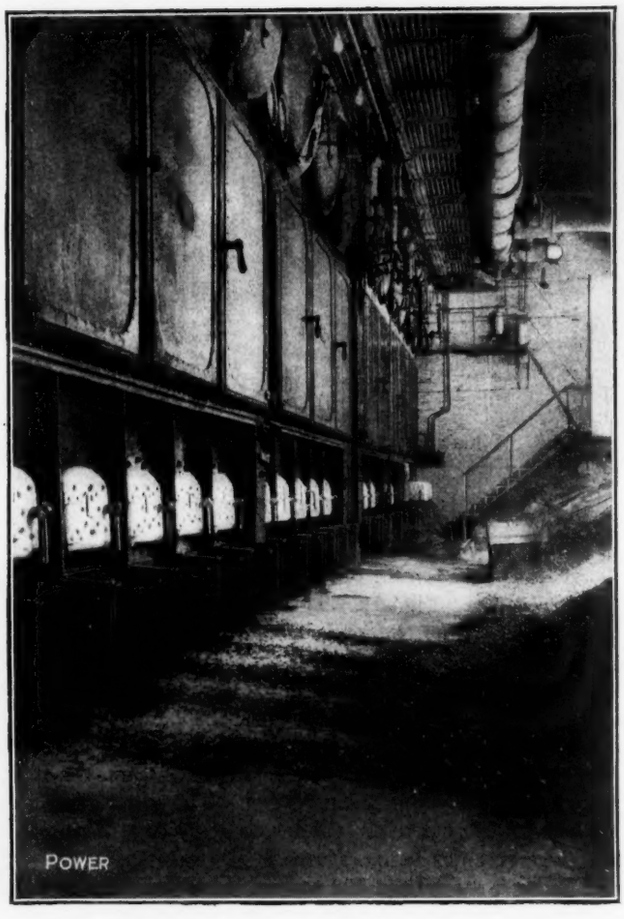


FIG. 7. ONE OF THE BOILER ROOMS, CONTAINING FIVE OF THE 14 BOILERS

of the boiler rooms, which are below the street level, but in the third room it is elevated by a bucket conveyor to a storage bin. Ashes are wheeled to an ash conveyor and elevated to a bin on the outside of the boiler house, from which they are loaded into wagons on the street level and carted away.

The pump room (Fig. 8) is a fair-sized steam plant in itself. The largest unit is the 16x42-in. engine, already mentioned. From the lineshaft which this engine drives there are belted a 5½x8- and an 8x10-in. triple-plunger house pump for supplying fresh water to the factory, washrooms, etc. Both are equipped with regulators which maintain a pressure of 90 lb. on the pipe system.

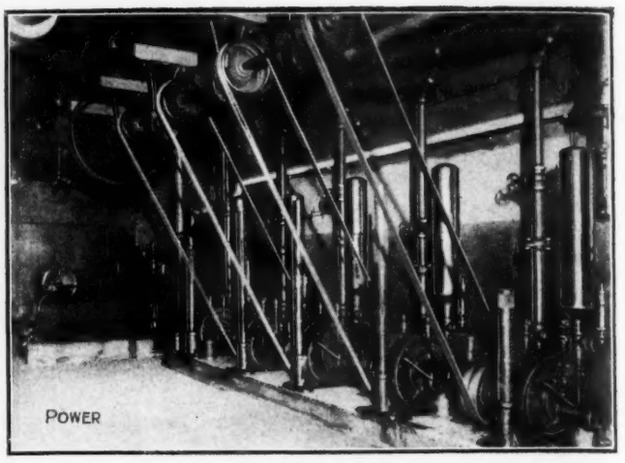


FIG. 9. BELT-DRIVEN HIGH-PRESSURE BLOWERS

Air for the factory is supplied by two compressors, one belt-driven from the lineshaft, the other being a compound steam-driven unit. Both supply air at 90 lb. pressure.

The factory has a system of vacuum cleaning. Vacuum is produced by two vacuum pumps, driven by a noiseless chain drive; each has a 24x18-in. cylinder and rated at 60 hp.

The refrigerating system not only keeps a proper temperature in the storage room for hat bodies, but it cools the drinking water for the factory. Since the introduction of this system of cooling the drinking water the rate of sickness among the workmen has greatly decreased. The system for cooling the brine supply consists of one 45-ton, 14x32-in. ammonia compressor, two 5x4-in. brine pumps, and the necessary apparatus. The drinking water is pumped to the factory by two 10x6x10-in. duplex steam pumps.

Boiler-feed water is supplied by a 14 & 20x10x18-in. and a 12x8½x12-in. duplex pump, working against 120 lb. pressure. Among the other apparatus is a 12x7x10-in. cold-water house pump, two vacuum pumps for the factory heating system, and a hydraulic pump for producing water pressures up to 300 lb. per sq.in. for the factory on various presses used in the process of manufacture.

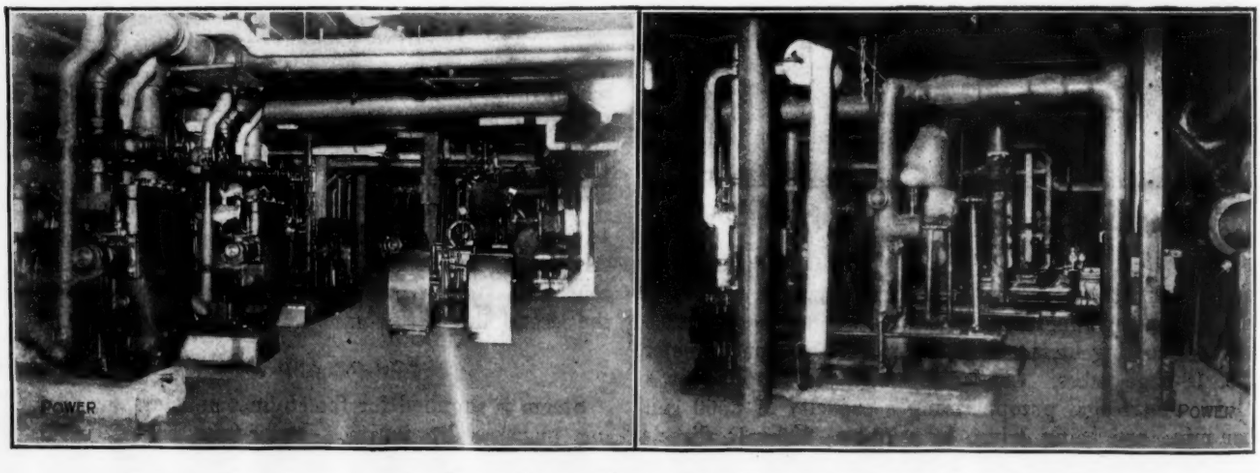


FIG. 8. TWO VIEWS OF THE PUMP ROOM, WHICH IS A FAIR-SIZED POWER PLANT IN ITSELF

At one end of the pump room is a set of five belt-driven, high-pressure blowers (Fig. 9) for the gas-heated irons in the factory. Each is fitted with a tight and loose pulley and is driven from an overhead shaft.

Miscellaneous

For the convenience of the employees a garage has been built, having steam heat, electric light, hot and cold water, compressed air, water for washing cars, asbestos and iron lockers, and a small machine shop for light repair work; also a charging station for electric automobiles. Out in the yard there is a rack for the accommodation of bicycles; they are stacked on end to occupy a minimum of room. A hospital is also maintained by the company and a large auditorium is available for entertainments.

The factory is wired with six trunk telephone lines, serving over 200 instruments, placed at convenient points. The 22 elevators are equipped with telephones, making it an easy matter for heads of departments to be reached when away from their desks.

A system of call bells has also been put in, so that no matter where the head of any department may be, the ringing of his signal denotes that his presence is required at his office.

PRINCIPAL EQUIPMENT OF THE JOHN B. STETSON CO. POWER PLANT

No.	Equipment	Kind	Size	Use	Operating Conditions	Maker
3	Engines	Reciprocating	24x48-in	Main units	90 r.p.m. saturated steam	Brown Engine Co.
2	Engines	Reciprocating	30x48-in	Main units	90 r.p.m. saturated steam	Brown Engine Co.
1	Engine	Reciprocating	8x8½-in	Used after work hours	Saturated steam	Westinghouse Machine Co.
1	Turbine	Rateau-Smoot	750 kw	Main unit	1500 r.p.m. exhaust steam	Ridgeway Dynamo & Engine Co.
3	Generators	Direct current	250 kw	Main generators	90 r.p.m., 230 volts, 800 amp.	Crocker-Wheeler Co.
2	Generators	Direct current	500 kw	Main generators	90 r.p.m., 230 volts, 2000 amp.	Crocker-Wheeler Co.
1	Generator	Direct current		Used after hours	250 volts	Westinghouse Elec. & Mfg. Co.
1	Generator	Direct current	750 kw	Main generator	1500 r.p.m., 230 volts, 3000 amp.	Ridgway Dynamo & Engine Co.
1	Engine	Corliss	16x42-in	Driving lineshaft, pump room	101 r.p.m.	Allis-Chalmers Co.
1	Condenser	Surface	4200 sq.ft. cooling surface	With turbine	27½-in. vacuum	C. H. Wheeler Mfg. Co.
1	Motor	Direct current	70 hp	Driving circulating pump	950 r.p.m., 220 volts	General Electric Co.
1	Pump	Centrifugal	12-in	With condenser	950 r.p.m.	C. H. Wheeler Mfg. Co.
1	Pump	Air Rotrex	18x36-in	With condenser	225 r.p.m.	C. H. Wheeler Mfg. Co.
1	Motor	Direct current	14 hp	Driving air pump	225 r.p.m., 220 volts	Sprague Electric Works
2	Heaters	Berryman	3000 hp	Feed water	Exhaust steam	Benj. F. Kelley & Sons
6	Boilers	Parker down draft	500 hp	Steam generation	Hand fired	Parker Boiler Co.
3	Boilers	Parker down draft	600 hp	Steam generation	Hand fired	Parker Boiler Co.
5	Boilers	Parker down draft	750 hp	Steam generation	Hand fired	Parker Boiler Co.
1	Pump	Triple plunger	5½x8-in	House service	Belt driven	Platt Iron Works
1	Pump	Triple plunger	8x10-in	House service	Belt driven	Platt Iron Works
1	Compressor	Simple		Factory use	90-lb. pressure	Ingersoll-Sargent Drill Co.
1	Compressor	Compound		Factory use	90-lb. pressure	Ingersoll-Sargent Drill Co.
2	Pumps	Vacuum	24x18-in	Vacuum cleaning system	Chain belt driven	Vacuum Cleaning Co.
1	Compressor	Ammonia	14x32-in., 45-ton cap.	Refrigeration	Steam driven	De La Vergne Mch. Co.
2	Pumps	Power plunger	5x4-in	Pumping brine	Belt driven	Fairbanks, Morse Co.
2	Pumps	Duplex	10x6x10-in	Cold drinking water	Steam driven	Henry R. Worthington Co.
1	Pump	Duplex, comp.	14&20x10x18-in	Boiler feed	Against 120-lb. pressure	C. H. Wheeler Mfg. Co.
1	Pump	Duplex	12x8½x12-in	Boiler feed	Against 120-lb. pressure	Advance Pump & Compressor Co.
1	Pump	Duplex	12x7x10-in	Cold water for factory		Henry R. Worthington
1	Pump	Vacuum	8x12x12-in	Heating system	During cold weather	Union Steam Pump Works
1	Pump	Vacuum	8x12x16-in	Heating system	During cold weather	Union Steam Pump Works
1	Pump	Hydraulic	12x9½x12-in	Factory hydraulic system	Steam driven	C. H. Wheeler Mfg. Co.
1	Pump	Hydraulic		Factory hydraulic system	Steam driven	Union Steam Pump Works
5	Blowers	High pressure	4 No. 3—1 No. 4	For gas	Gas heated irons	American Car Furnace Co.
1	Tower	Forced and natural draft	27x27x75-ft	Cooling condensing water	Forced and natural draft	C. H. Wheeler Mfg. Co.
4	Fans	Forced draft	8 ft	Cooling towel	Motor driven	C. H. Wheeler Mfg. Co.
4	Motors	Direct current		Driving tower fans	220 volts	General Electric Co.

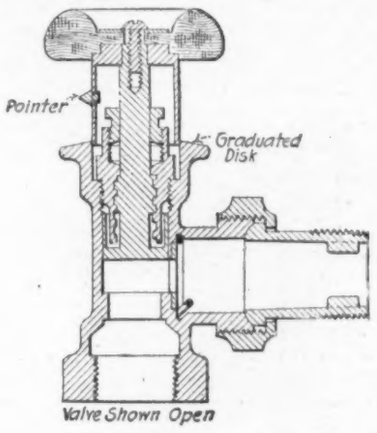
In connection with the engineering staff, Chief Engineer D. J. Wattis also has supervision of the machine shop, the force totaling 140 men.

Adsco Graduated Radiator Valve

The improved Adsco radiator valve, illustrated herewith, is calibrated to supply a definite amount of radiation, and is designed to supply steam to standard sizes of radiators of the hot-water type.

The various capacities of valves are arranged in multiples of five square feet of direct radiation. These capacities have been established as the result of tests to determine the ratio of the amount of steam under a given pressure passing through the radiator valve, to the amount of steam condensed per square foot of radiation under average demand conditions.

When used in connection with the atmospheric system



Section through Atmospheric Radiator Valve

of steam heating, it affords control of individual radiators, and just the amount of steam needed is used in each radiator to maintain the desired temperature.

The valve body is fitted with a graduated disk at the outlet. The valve travels from a closed to a full open position with a three-quarter turn. A pointer on the valve stem indicates on the graduated disk any fractional position between the two extremes. The claims for this valve, manufactured by the American District Steam Co., North Tonawanda, N. Y., are economy, freedom from clogging and that it will not stick after remaining idle.

A Common Error when trouble appears in the form of scored cylinders and valves is to hold the oil responsible for the damage. After the necessary repairs are made and possibly the mechanical cause of the trouble removed, another oil is substituted with very satisfactory results. Both oils may have come from the same field, had the same composition and physical properties, in fact, be the same oil, but from barrels with different trade names, so that the admiration for the second, together with the condemnation of the first, would be unjustified.

Lauson Heavy-Duty Kerosene Engine

Recently the John Lauson Manufacturing Co., New Holstein, Wis., placed on the market a four-cylinder, vertical, heavy-duty oil engine, of 80 and 100 hp., with cylinders 10¼x12 in. and 11x12 in., respectively. Primarily, the engine is intended for small lighting-plant work, the regulation being close enough to permit operating alternating-current generators in parallel.

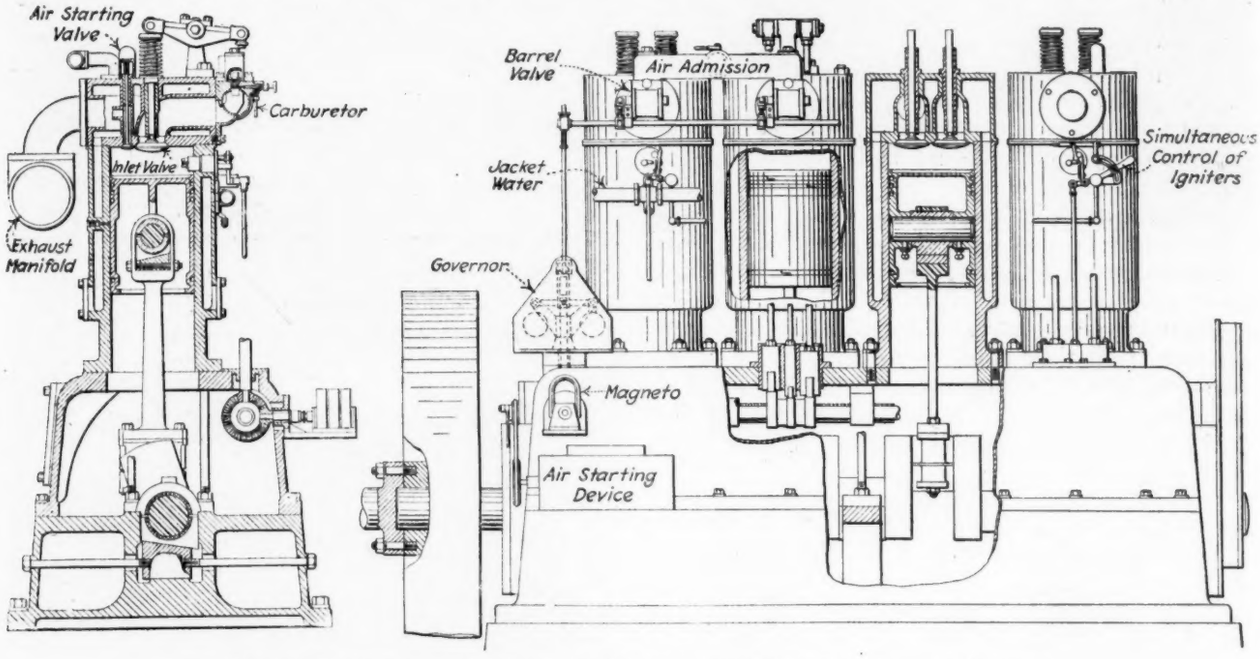


FIG. 1. TRANSVERSE SECTION AND LONGITUDINAL VIEW OF LAUSON ENGINE

The general construction is shown in Fig. 1. The crank case is of the two-piece type split horizontally at the center of the shaft. The valves are mechanically operated and located in the head. Make-and-break ignition is employed and cooling water for the cylinder jackets is supplied by a pump driven through a chain and sprocket by the main shaft.

The feature of the engine is the special carburetor or "Venturia" mixing nozzle, of which there is one for each cylinder. The principle of this device is to maintain a high velocity of air through a venturi tube having radial holes in its restricted portion through which the fuel is drawn by the suction of the air. The amount of air passing through the nozzle is controlled by the governor. The carburetor consists of a cast-iron body containing a cylindrical throttling chamber within which is fitted the barrel valve shown at *F*, Fig. 2. This valve is controlled by the governor through a bell crank and a horizontal shaft running the length of the four cylinders.

Fuel is admitted through the nozzle *D*, which has two sets of holes. The upper set is for the admission of gasoline for starting and kerosene for running, and the lower set for the admission of water to prevent premature ignition at full load. Three needle valves, one for each liquid, control the supply to the nozzle. On each side of the nozzle *D* is a port *I*, and when the engine is at rest these ports are wide open. Below the nozzle the governing port *P* will also be open. Thus, a certain proportion of the air passes through ports *I* and the balance through the nozzle *D*. As soon as the engine picks up in speed, valve *F* is rotated by the governor, gradually closing the ports *P* and *I*, so that a greater proportion of the air is deflected through the nozzle, thus maintaining a practically uniform velocity at this point The high velocity of the air and the feeding of the fuel through a number of small holes insure atomization and a proper mixture before the fuel passes to the cylinder. A butterfly valve controlled by the handle *E* is provided to facilitate starting and for additional air adjustment at full load.

The igniters are of standard make-and-break design and operated from the camshaft. Two timing adjustments are provided, one individual and one simultaneous, the latter being used to shift all igniters at times of starting by a single lever. The mechanism is shown in Fig. 1. An insulated brass bar charged by a gear-driven alternating-current magneto is placed above the igniter plug. A spring coming in contact with the bar puts the igniter in the circuit and eliminates the need for wiring.

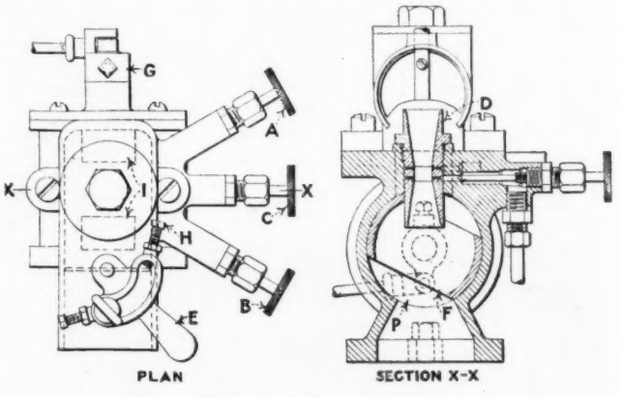


FIG. 2. THROTTLING CHAMBER

For starting, a special device furnishes the air in turn to each cylinder. The device consists of a main body having four radial air ports connected by piping to the different cylinders. These ports are covered and uncovered by a rotary disk valve having one port. The valve is

held to its seat by the pressure of the air and is free to revolve when the air is shut off. Rotation is effected by means of a flexible coupling between the device and the camshaft. Compressed air, provided in the usual way, is admitted to the cylinder through a small valve in the head, which is shown at the left in the sectional view of the engine. As soon as the engine fires, this valve is held to its seat by the pressure within the cylinder. The engine is run for about ten minutes on gasoline and the fuel is then changed to kerosene.

The fuel reservoir has three compartments; one for gasoline, one for kerosene and one for water. The gasoline and kerosene compartments are kept full by means of pumps, and the water is controlled by a float inside the chamber.

The governor is of the vertical flyball type driven from a bevel gear on the camshaft. It is inclosed, as indicated in Fig. 1, and the speed may be adjusted by shortening or lengthening the rod from the governor to the regulating valve under its control. Lubrication of the five main bearings and the cylinders is effected by a force-feed pump. The connecting-rods depend on splash lubrication.

Quarter-Turn Rod Coupling

To obviate the noise and wear of a bevel gear between a jackshaft belt driven by a motor set on the engine-room floor and a centrifugal pump some distance below, C. P. Hall, chief engineer of the Rookery Building, Chicago,

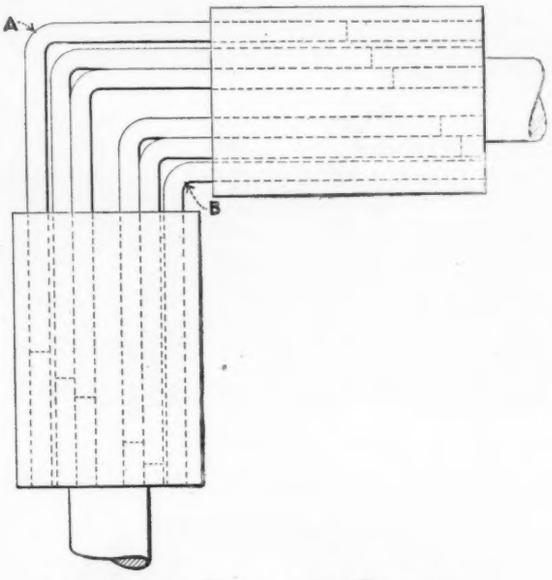


FIG. 1. DIAGRAM OF COUPLING

invented an ingenious quarter-turn coupling consisting of two heads, bored to receive six rods of equal length. The jackshaft is horizontal and the pump shaft vertical, as indicated in Fig. 2. The coupling heads are merely solid pieces, bored for and keyed to their respective shafts. Within one-half inch of the circumference and spaced evenly around it, six holes are drilled to comfortably receive the rods. These rods are twice the length of a head plus the shortest exposed length shown in Fig. 1. They are free to turn in their sockets and slide lengthwise as the relative movements of the heads demand. When in the extreme position *A*, the ends of a rod are midway in the heads and in position *B* the ends are flush with the outer faces.

At first glance it would look as though the rods would twist together in a single turn of the heads. That this is not the case has been successfully demonstrated by Mr. Hall. Several couplings of the same kind as illustrated are in use in his plant. Fig. 2 shows a coupling connecting a jackshaft, driven by a 3-hp. motor, to a 3-in. centrifugal pump. The speed is 500 r.p.m., and at a dis-

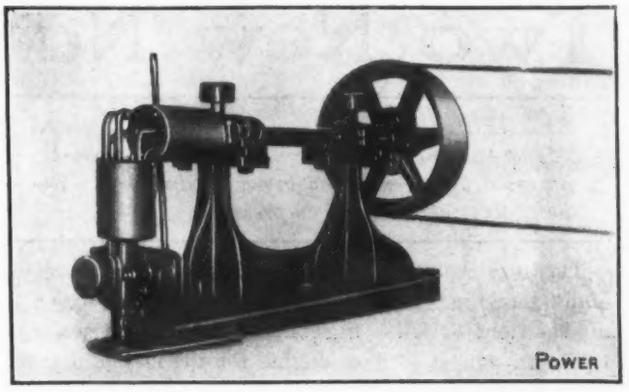
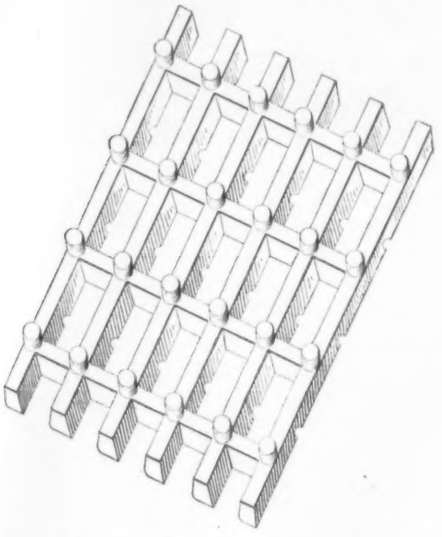


FIG. 2. JACKSHAFT AND COUPLING WITH HOOD REMOVED

tance of only a foot it was impossible to detect any noise from the coupling. The rods are well greased and as a precautionary measure a hood is placed over the coupling. The largest coupling is for a 10-hp. motor, but there is no reason why higher powers could not be transmitted; it is merely a question of size. To be safe the combined area of three rods should be equal to the area of the shaft.

Keystone Grease Retarder

On large bearings where grease is used for lubrication, if the grooves are not cut in line with the center of the bearing, part of the journal is without lubrication. Retarders made from copper-wire gauze, perforated copper



KEYSTONE GREASE RETARDER

plates, perforated wood or leather have been used with grease, but none of these have been entirely satisfactory.

It is claimed that the difficulties usually experienced in lubricating such bearings have been overcome by the use of the Keystone babbitt-metal retarder, illustrated

herewith. This retarder is bent to conform to the curve of the journal and is made slightly narrower and shorter than the grease well; the grease is placed on top of the retarder in the usual manner.

The under side of the retarder is grooved, and one edge of each bar is rounded, so that the grease is wedged between the retarder and the journal instead of being scraped off. The grease is spread over the bearing surface and fed into the groove in the bearing cap, providing efficient and economical lubrication.

This device was designed by Thomas O. Organ, consulting engineer of the Keystone Lubricating Co.

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Two New Nordberg Engines

SYNOPSIS—Cylinder a plain casting. Balanced oppet valves contained in heads. Valves positively operated by eccentric on lay-shaft and cam. Relief valves avoid over-compression.

Two new types of engine being built by the Nordberg Manufacturing Co., of Milwaukee, Wis., are shown in the shop photographs, Figs. 1 and 2. The former is an 18x32-in. poppet uniflow engine for the city of Bartow, Fla. It will drive directly a 150-kw. alternator at 164 r.p.m. Fig. 2 shows an 18x24-in. poppet valve "counterflow" engine which will be directly connected to a 175-kw. alternator and run at a speed of 164 r.p.m. Waupun, Wis., is to have this unit. All but the lower half of the flywheel on each engine is assembled. The simplicity of design will be apparent, especially that of the uniflow engine.

The word "counterflow" is used to make clearer the distinction between the uniflow type and an engine into which the steam flows in the ordinary way. So many new engines have been perfected within recent years that there is some confusion as to what the different names mean. The word "uniflow" has been coined to designate an engine in which the steam flow within the cylinder is only in one direction—unidirectional flow. In the usual engine there is a reversal of steam flow; on the outstroke the flow is toward the advancing piston, and on the return stroke the same steam flows toward the cylinder head. In the uniflow engine the steam is admitted at the ends, as in the ordinary engine, but is exhausted through ports at the center of the cylinder, the piston acting as its own exhaust valve, as shown in Fig. 3.

The uniflow principle has to do with only the cylinder and exhaust-valve design, so that an engine of this type may be fitted with any style of valves and valve-gear for controlling the steam inlet. Corliss valves may be

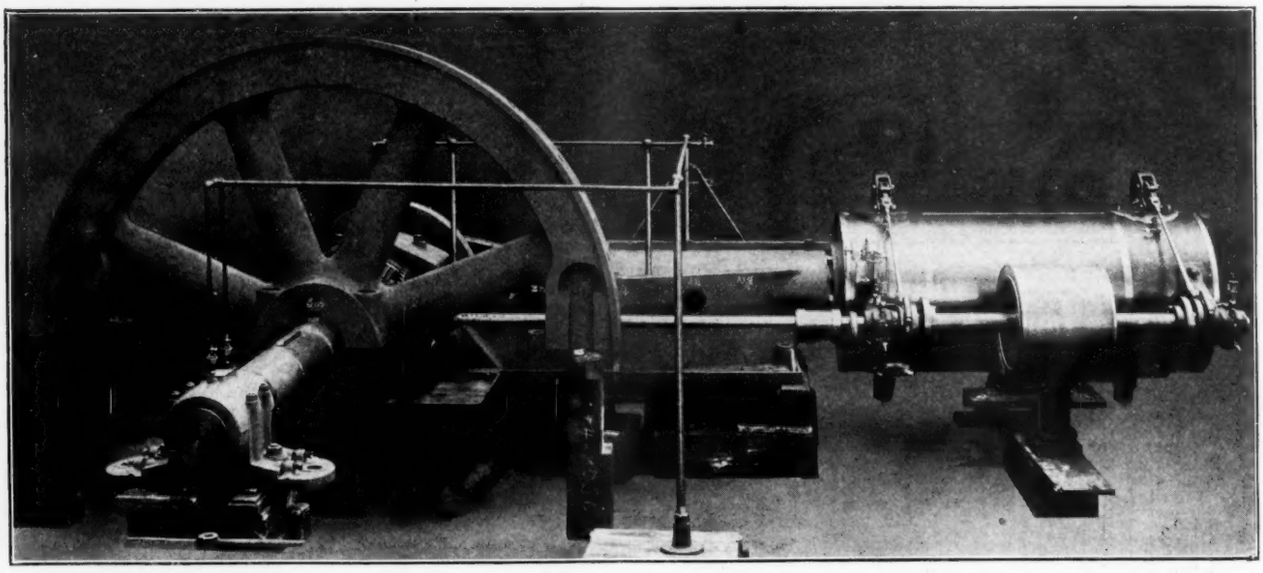


FIG. 1. NORDBERG POPPET-VALVE UNIFLOW ENGINE, 18X32-IN. CYLINDER

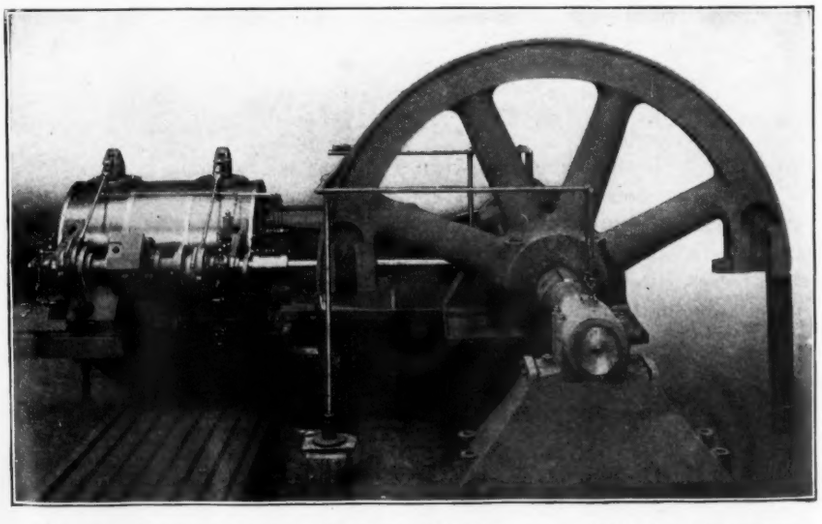


FIG. 2. POPPET-VALVE COUNTERFLOW ENGINE, 18X24-IN. CYLINDER

used, and a number of Nordberg uniflow engines have been so equipped. A description of this engine appeared in the June 11, 1912, issue of POWER. For high pressures and superheats, however, poppet valves are to be preferred.

Line drawings of both of the new engines are shown in Figs. 4 and 5. The frame is the standard Nordberg

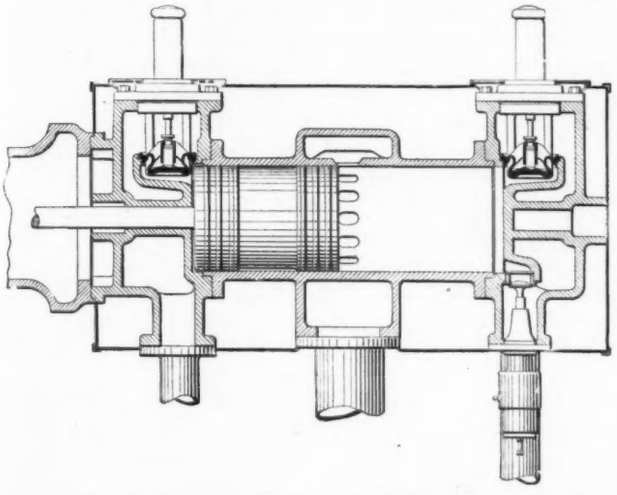


FIG. 3. SECTIONAL VIEW OF UNIFLOW CYLINDER

heavy-duty design with an oil pan cast integral under the crank and rod. The bearing, rods, guides and crosshead are also standard. The cylinders of these engines are of plain cylindrical form without steam chests—this to avoid distortion under high superheat. The steam

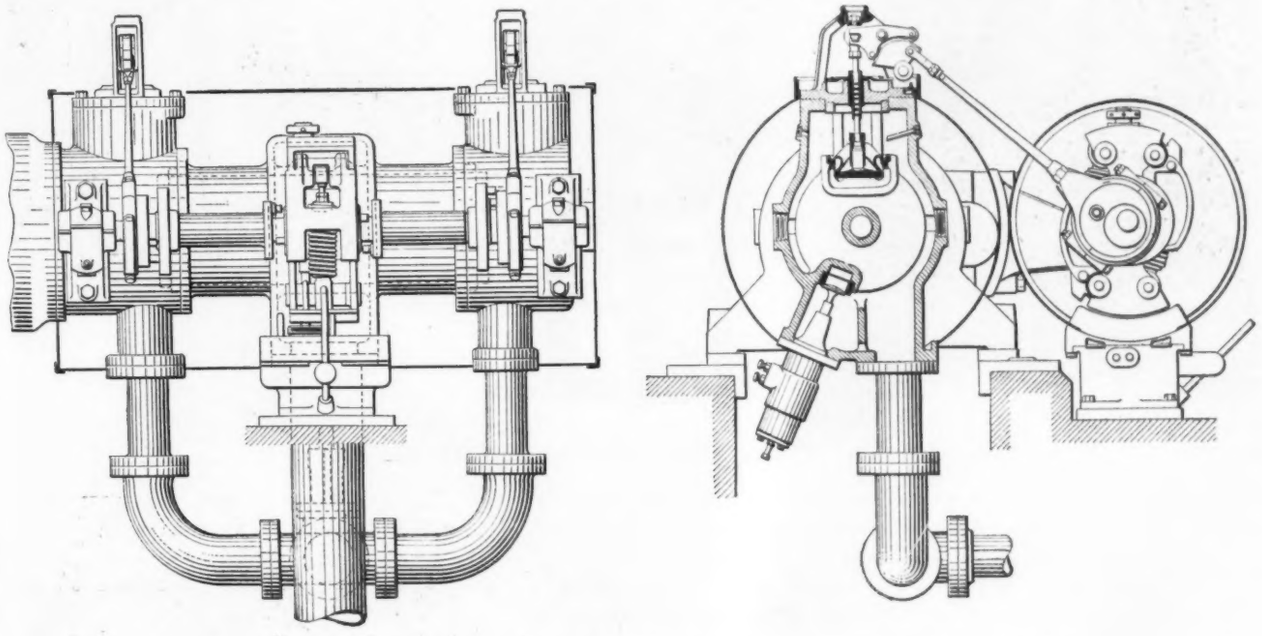


FIG. 4. LONGITUDINAL VIEW AND TRANSVERSE SECTION OF UNIFLOW CYLINDER

is led to each valve separately from the throttle valve placed under the floor. The cylinder heads are cast separately and contain the valves, and, as shown in Fig. 3, the design is such that the entering steam jackets the ends of the cylinder.

The arrangement of the cylinder and heads is shown in Fig. 6. To dismantle, the head with the valves is removed by backing off on the rack. The cylinder may then be removed from the crank-end head. The valves are of the double-beat balanced poppet design, shown together with the operating cam and follower in Fig. 7. The valves seat on removable cages, which are steam tight in the cylinder-head casting. This construction was adopted to obviate the distortion common to seats cast integral with the cylinder casting. These cages can be renewed or removed for regrinding. No stuffing-boxes or metallic packing are used on the valve stems. These are ground to a close fit and then made tight by grooves which prevent leakage, on the principle of the labyrinth used in centrifugal pumps, compressors, etc.

The stubby, compact appearance of the valve bonnets is due to the absence of springs for closing the valves. In this construction the valve is opened and closed positively by one cam oscillated by an eccentric on the lay shaft, the throw of which is varied by an inertia and centrifugal governor located between the eccentrics. The design of cam, eccentric and governor is shown by the illustrations.

In the counterflow type of engine there are four cams—two at each end—one for the steam inlet, the other for the steam exhaust valve. In the poppet uniflow engine there are only two cams—one for each steam-inlet valve—the exhaust, as already explained, being controlled by the piston itself.

The uniflow engine is primarily a condensing engine. Expansion may be carried from high boiler pressure to 26 in. of vacuum within one cylinder at as good economy as ordinarily obtained with a compound condensing engine, owing to the reduction in cylinder condensation effected by the uniflow construction. This type of engine has the further advantage of large overload capacity and a flat steam curve. It is claimed that it will carry 100 per cent. overload with a 10 per cent. increase in steam per horsepower-hour over the full load rate. At half load the steam rate is about 5 per cent. in excess of normal.

An objectionable feature of the uniflow engine is the high compression obtained when the vacuum is lost or when the engine is run noncondensing. In the present design this difficulty has been met by placing an automatic relief valve at each end of the cylinder. One of these valves is shown in Fig. 4. It opens from the clearance space and discharges the steam in its superheated state at the end of compression back to the steam piping; overcompression is thus avoided.

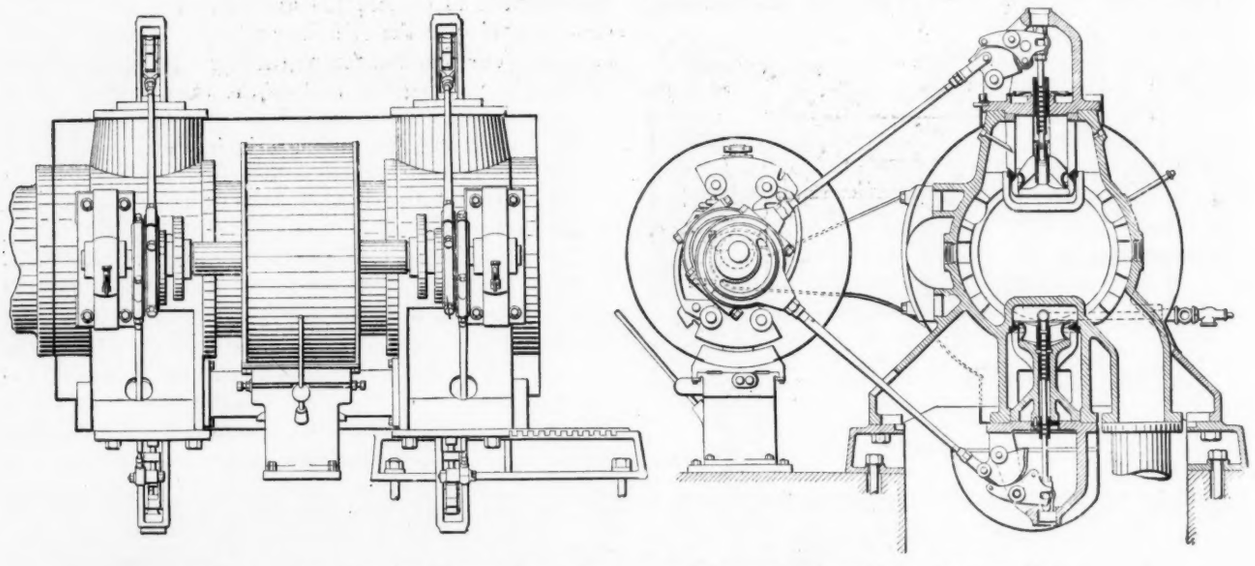


FIG. 5. THE POPPET-VALVE CYLINDER WITH POSITIVE HIGH-SPEED VALVE-GEAR

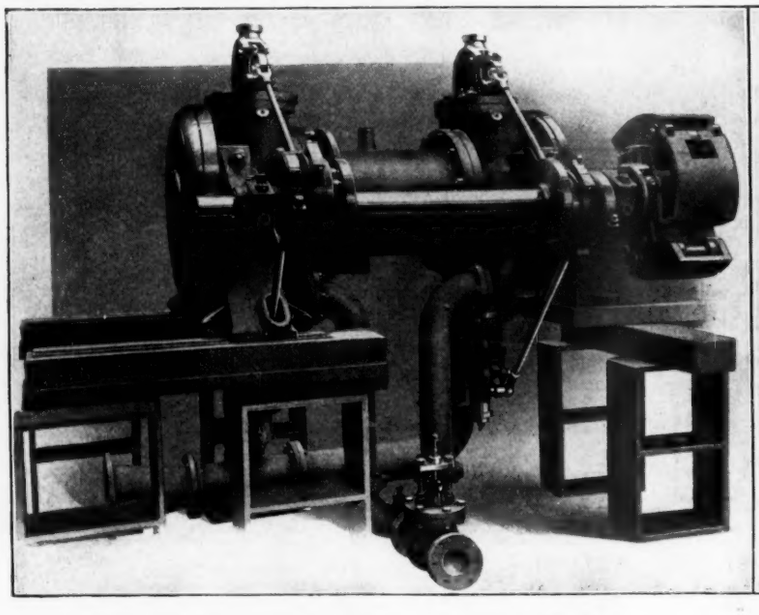


FIG. 6. CYLINDER AND HEADS, BEFORE COVERED BY LAGGING, AND VALVE-GEAR

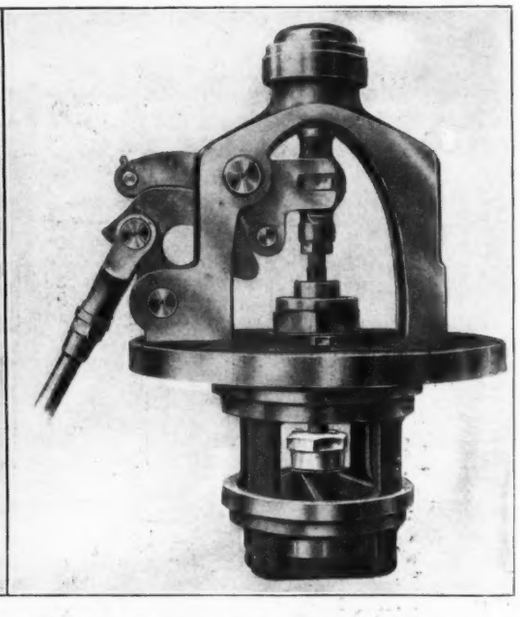


FIG. 7. BALANCED POPPET VALVE WITH OPERATING CAM AND FOLLOWER

Pumping Water with Compressed Air—A 12x14¼x14-in. compressor furnished air for a mine pump 14x8x3 in. No other uses were made of the air and the air line was tight. Indicator cards were taken from both the air and steam cylinders of the compressor. The valve adjustments were good and the pistons tight. The total pumping head of the pump, including suction and pipe friction, was 103.1 ft. The water pumped was measured by a 4-in. orifice in a tank at the surface. The over-all efficiency from steam indicated horsepower to useful work done on the water was only 6.81%.

Crankpin Troubles

By C. W. Haynes

One of the common troubles encountered in the engine room is the undue heating of crankpins. Not properly relieving the brasses at the parting is one cause, and another is in not allowing sufficient clearance at the fillets.

A brass with no clearance may run for some time with little or no trouble, but after heating has taken place, it will be found upon taking it down that it has developed a decided tendency to grip the pin. To avoid this the clearance should be ample to reduce the bearing on the brass to the crown of the pin.

In roundhouse work, in a locality where the road was hilly, it was the common practice to reduce the area to nearly one-half of the arc of the brass without bad results. It would seem that this excessive clearance would

result in heating, but it did not, even when the engines were pulling hard and running at high speed, or going down hill with the steam shut off, at which times the thrashing of the rods tries the pins severely. This shows that seemingly excessive clearance will not cause heating.

The locomotive engineer has troubles that the stationary engineer does not experience. In dry times the wind blows the dust, and in wet weather the mud between the ties is thrown up into the bearings; then when the oil is used up, out goes the babbitt. If the babbitt is all thrown out, there can be but little injury, but if the engine is stopped and the partly melted babbitt is allowed to freeze on the pin, cutting is liable to occur.

It is essential that the brasses be properly fitted and no high spots left to cause severe local heating. All of the flaky substance which covers a brass after heating should be filed or scraped away. It is not necessary, however, to scrape away all the file marks, which, being crosswise, afford lodgment for oil and are beneficial in newly fitted bearings.

A soft babbitt gives satisfaction when used in the inserts, but it should be peened in or it will get loose. Good results have been secured by fitting the brasses first and babbitting afterward. In doing so it is easy to leave the babbitt a little higher than the crown of the brass, insuring the benefit of the good qualities of the babbitt.

The following experience illustrates the benefits of babbitt inserts in bronze bearings. Certain gasoline engines using phosphor bronze on the crankpins gave trouble frequently until we put two rows of button inserts across each brass. Later, the use of phosphor bronze was discontinued in favor of a high-grade antifriction metal.

Not all the trouble with crankpins, however, is due to mechanical faults. The quality of the oil is responsible in many cases of heating. The superintendent one day wanted to know the cause of the poor condition of the crankpin bearings of a two-cylinder engine. He said the engine must have run dry earlier in the day. It was contended, however, that if the engine had run dry, the metal would have melted, but with a poor grade of oil the bearing had heated gradually, softening the metal and allowing it to crush out at the sides. That same grade of oil did give much trouble later.

When a newly fitted brass is put in hard service, a little tallow in the clearances helps to prevent heating. Fast-running engines must be kept snugly keyed up or the brasses and straps will chafe. When straps are off it is advisable to look them over for cracks, especially in the corners, where they are often easy to see. It is, however, good practice to wipe them carefully and paint with a thin coat of white-lead paint, then strike the strap or rod end with a soft hammer and if there are any cracks the oil in them will discolor the paint, thus indicating the extent of the fracture.

Brasses running loose at high speed will pound themselves hot and may fracture the running parts.

Forced-Draft Cooling Towers

By E. Raymond Goodrich

SYNOPSIS—Analyses and test data of forced-draft cooling towers, with heat-temperature curves for moist air, and examples illustrating their use.

In a rapidly increasing number of condenser installations, some sort of water-cooling medium becomes a necessity, and, on account of its small space requirements, the forced-draft cooling tower is generally the most feasible. Since apparently little is known of the theory involved and also the practical limitations of the problem, it is the writer's intention to lay down certain principles, substantiated by accurate experimental data, which will enable one to choose intelligently between different designs and estimate the quantities involved.

The physics of water cooling is comparatively simple, and for present purposes is best illustrated by reference to Fig. 1, which represents a typical vertical section of an inclosed tower. Briefly, the operation is as follows:

Hot water enters the tank *a*, is broken up into streams as it leaves the distributing troughs at *b*, and trickles down through the filling *c* where it is still further broken up and comes into contact with the up-going air which enters at *d*. The cooled water is withdrawn from the reservoir *e*.

Inclosed towers are divided into two classes, forced and natural draft, according as the air is forced in at *d* by fans or flows in naturally due to the upward chimney effect when a tall stack is used. The types of towers on the market are identical with the section given in Fig. 1, the distinguishing feature of different manufacturers being merely the arrangement of the distributing system and the kind of filling used. The following analysis holds good for all types.

Water passing through a cooling tower gives up its heat in three ways: First, by radiation through the walls of the tower; second, by direct contact with the up-going air; and third, by evaporation of a part of the water to be cooled. The first item is so small as to be well within the errors of any test and consequently may be regarded as negligible. The amount of cooling due to contact with the out-going air will now be considered. Call this h_1, in B.t.u. per minute. Then

$$h_1 = W S \ (t_2 - t_1);$$

where

W = Pounds of air passing through per minute;
S = Specific heat of air at constant pressure (0.2375);
t_2 = Temperature of the outgoing air;
t_1 = Temperature of the incoming air;

or, as it will be more convenient to express the air in thousands of cubic feet per minute,

$$h_1 = QK \ (t_2 - t_1) \qquad (1)$$

Where Q is the quantity of air in thousands of cubic feet per minute, and the factor K represents the amount of heat required to raise the temperature of 1000 cu.ft. of air through 1 deg. F., taking into account that as the temperature rises, a pound of air increases correspondingly in volume. This value is given by the curve marked K, Fig. 2, and should be taken at the temperature of the outgoing air.

The heat loss represented by h_1 is between 15 and 20 per cent. of the total cooling, depending on atmospheric conditions.

Next consider the third item, the amount of heat abstracted by evaporation of part of the entering water, which represents by far the greater part of the cooling. Obviously, since all the heat lost by the water must be carried away by the air, the amount of this evaporation and consequent cooling will be limited only by the moisture-carrying capacity of the air. A cubic foot of air can contain only a certain amount of moisture, depending on its temperature, and when this maximum condition obtains, the air is said to be saturated at that temperature. When the air actually contains a smaller amount than this, then the ratio of this amount to the maximum possible amount is called the relative humidity, expressed in per cent. When air contains its maximum moisture at any temperature—that is, the humidity is 100 per cent.—if the temperature is decreased moisture will be precipitated; on the other hand, if the temperature is raised, the air will no longer be saturated but will be capable of absorbing a certain additional quantity of water vapor. It is this increasing moisture capacity with increasing temperature that has made the inclosed cooling tower the most efficient means of water cooling; in it the air is heated to the highest possible temperature, approaching that of the inlet water, and consequently evaporates the maximum quantity of moisture. The curve marked *P*, Fig. 2, gives the vapor-carrying capacity of air in pounds per thousand cubic feet at 100 per cent. humidity for different temperatures, the moisture content at any other degree of humidity being obtained by taking the corresponding percentage of these values.

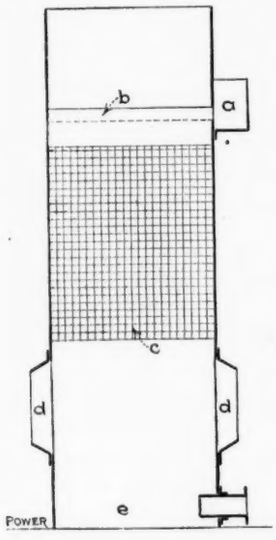


FIG. 1. DIAGRAM OF TYPICAL COOLING TOWER

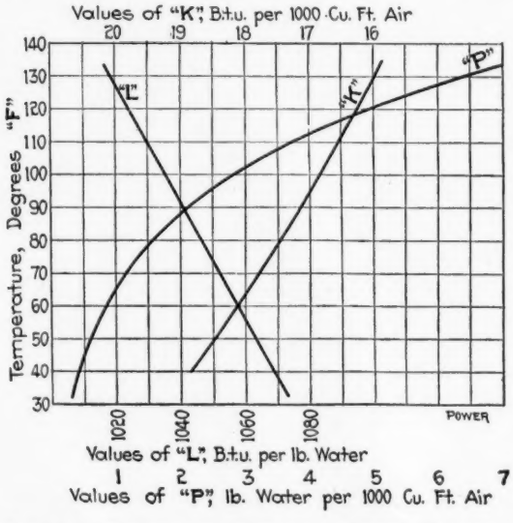


FIG. 2. VALUES OF K, L AND P WITH VARYING TEMPERATURES

Atmospheric air entering a cooling tower is rarely saturated, its degree of humidity being readily obtainable by taking simultaneous readings with the "wet-and-dry-bulb" thermometer and referring to tables in any engineering handbook; but as all cooling-tower estimates are made with direct reference to the humidity and proposals and guarantees are based on this factor, we will not extend our calculations further than this as a starting point. Calling the heat lost by evaporation h_2 and remembering that all the heat lost by the water is gained by the air, the general equation of heat transfer in a cooling tower becomes

$$H = h_1 + h_2 \qquad (2)$$

where,

$$H \text{ (\textit{heat lost by the water})} = 8.3\, G\, (T_2 - T_1) \qquad (3)$$

G being the gallons per minute passing through the tower, and T_2 and T_1 the respective temperatures of the incoming and outgoing water.

In order to best illustrate this, the analysis will be applied to test No. 1 in the table of "Tests on a Forced-draft Cooling Tower." The quantities are: $G = 651$; $T_2 = 105$ deg.; $T_1 = 84.7$ deg.; $t_2 = 90$ deg.; $t_1 = 71$ deg.; humidity in = 40 per cent.; humidity out = 100 per cent.

Therefore,

$$H = 8.3\ (651 \times 20.3) = 110{,}000 \textit{ B.t.u. per min.}$$

The B.t.u. by direct heating will be

$$h_1 = QK\ (90 - 71)$$

where Q is the quantity of air in thousands of cubic feet per minute, and K is taken at 90 deg., equal to 17.15 (see Fig. 2). Therefore,

$$h_1 = 17.15 \times 19 \times Q = 326\, Q \qquad (4)$$

The term h_2 is obtained as follows: The moisture content at 71 deg. and 40 per cent. humidity will be $1.16 \times 0.40 = 0.464$ lb., which is the amount of moisture entering the tower with every thousand cubic feet of air. On leaving the tower, the entering air will have increased in volume due to the rise in temperature, so that the quantity of air leaving per minute will be

$$Q\left(\frac{550}{531}\right) = 1.034\, Q$$

where 550 and 531 are the absolute temperatures of the outgoing and incoming air. The moisture content of 1000 cu.ft. at 90 deg. and 100 per cent. humidity is 2.13 lb., the humidity of the outgoing air always being approximately 100 per cent. in a properly designed tower. As there is $1.034\, Q$ thousands of cubic feet leaving the tower, the moisture carried away by the original air that entered the tower will be $2.13 \times 1.034\, Q = 2.205\, Q$. Subtracting from this the moisture held by the air as it entered, gives

$$M = 2.025\, Q - 0.464\, Q = 1.561\, Q \qquad (5)$$

which is the amount of moisture actually evaporated from the water going through the tower, and represents the quantity of makeup water to be supplied per minute.

In order that water may evaporate, it must absorb a certain amount of heat per pound evaporated, called the latent heat of evaporation, the value of which depends upon the temperature at which evaporation takes place; this is given by curve *L*, Fig. 2. As all the moisture is finally heated up to the temperature of the outgoing air, it is correct to use the value of L at this tem-

perature. Therefore, the heat absorbed due to evaporation by the air leaving the tower will be $h_2 = 1.561\ Q \times 1041 = 1628\ Q$.

Now by equation (2),

$$H = h_1 + h_2$$

Substituting:

$$110{,}000 = 326\ Q + 1628\ Q$$

and,

$$Q = \frac{110{,}000}{1954} \times 1000 = 56{,}294\ \textit{cu.ft. per min.}$$

By referring to the test, it will be seen that 53,900 cu.ft. of air was actually measured by the anemometer, which is

RESULTS OF TESTS ON A FORCED-DRAFT COOLING TOWER, AVERAGES OF THREE-HOUR READINGS

Gal. per Min.	Temp. In, Deg. F.	Temp. Out, Deg. F.	B.t.u. per Min.	Temp. In, Deg. F.	Humidity In, Per Cent.	Temp. Out, Deg. F.	Humidity Out, Per Cent.	Quantity Cu.Ft. per Min. Measured by Anemometer	Quantity Cu.Ft. per Min. Calculated
651	105	84.7	110,000	71	40	90	100	53,900	53,000
638	107.8	87.5	108,000	72	60	93	100	50,100	51,000
638	112	88.5	124,500	66	60	96	100	51,400	49,400
643	108.5	87	115,000	69	48	92	100	50,200	49,000
640	109.9	90.5	103,400	83	48	95	100	50,600	51,800
*632	116	98	94,800	43	75	101	100	23,500	24,500
*630	135	115.8	102,000	60	73	118	100	17,575	18,250

*Natural draft, fan not running.

NOTE—These tests were conducted with utmost regard as to the accuracy of measurements. The quantities of air per minute were obtained by anemometers being moved back and forth across the top of the tower at regular intervals, and the results were corrected so as to give the actual amount entering the tower. The water was measured both by a venturi meter and a calibrated pitot tube.

or about 10.5 gal. per min. of makeup water must be supplied to compensate for the loss by evaporation.

In order to greatly lessen the foregoing calculations, the "heat-temperature curves for moist air" (Fig. 3) have been compiled with special reference to cooling-tower work. A few remarks on these are necessary. The ordinates represent Fahrenheit temperatures and the abscissas total heat content in B.t.u. per thousand cubic feet of air. This heat content by no means represents an exact "heat potential" at any temperature, and it is only the difference of any two values which has a physical meaning. Neither are the quantities absolutely correct from a theoretical standpoint, as certain factors have been omitted to make the calculation possible; but they will give results which are correct to within 4 or 5 per cent., which is close enough for this class of work. To illustrate the use of the chart, again assume the conditions of test No. 1 in the table. As before, the heat H lost by the water equals 110,000 B.t.u. per min. From the curves the heat content per thousand cubic feet at 90 deg. and 100 per cent. humidity is 3350 B.t.u. The heat content at 71 deg. and 40 per cent. humidity is 1240 B.t.u. Therefore, 3340 — 1250 = 2090 B.t.u. is absorbed per thousand cubic feet passing through the tower, and

$$Q = \frac{110{,}000}{2090} \times 1000 = 52{,}631\ \textit{cu.ft. of air per min.}$$

By simply reversing the process, the terminal temperature may be found if the quantity of air is known. The column marked "Quantity Calculated" in the table was figured

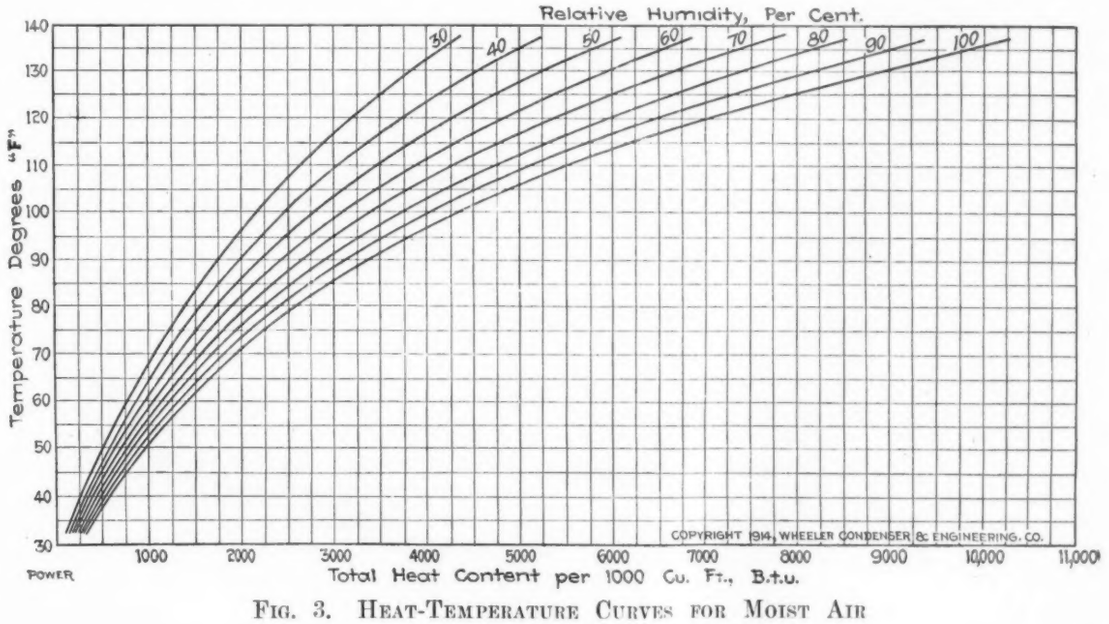


FIG. 3. HEAT-TEMPERATURE CURVES FOR MOIST AIR

a variation of only 4 per cent., or as close as might be expected, considering the difficulties involved in this kind of measurement.

By substitution of Q in (4) and (5), we have

$$h_1 = 326\left(\frac{56{,}200}{1000}\right) = 18{,}321\ \textit{B.t.u. per min.}$$

This is 16.7 per cent. of the total 110,000 B.t.u. given up by the water; that is, approximately 83.3 per cent. of the cooling is due to the evaporation effect alone. Also,

$$M = 1.561\left(\frac{56{,}200}{1000}\right) = 87.5\ \textit{lb.}$$

from these curves and shows how closely they check with actual results under a wide variety of conditions.

The maximum temperature of the outgoing air is limited by the temperature of the incoming water, and from an examination of curve P, Fig. 2, it is seen that as air increases in temperature, its water-carrying capacity, which represents about 85 per cent. of the total cooling, increases very rapidly. Due to this enormous increase in water absorption at the upper part of the temperature scale, it is evident that the air should leave the tower as near as possible to the temperature of the inlet water, and that an exact knowledge of this terminal difference

is essential in figuring the amount of air required for a given duty. In fact, with other conditions remaining the same, the ratio of the temperature of the outgoing air to that of the incoming water represents the real efficiency of any cooling tower. A short calculation will bear this out. Suppose that the air in test No. 1 left at 5 deg. below the incoming water temperature—that is, at 100 deg.; then,

heat content at 100 *deg. and* 100 *per cent. humidity* = 4350 *B.t.u.*

heat content at 71 *deg. and* 40 *per cent. humidity* = 1240 *B.t.u.*

The difference is 3110 B.t.u. Therefore, the air required will be

$$\frac{110{,}000}{3110}(1000) = 35{,}369 \textit{ cu.ft. per min.}$$

as against 52,631 cu.ft. with 15 deg. difference. This means a saving of 34 per cent. in fan power as well as a much lower air velocity, lessening the tendency to cause spray, which is sometimes very objectionable, as well as wasteful. The terminal difference of 15 deg. obtained in the test is by no means average practice, and the results are given merely to show the close agreement of theory and practice in calculating air quantities. In later experiments, made with a view to increasing the efficiency, the quantity of air was not measured. Just how successful these experiments have been is demonstrated by the report of a test recently made on a large installation in the Middle West, consisting of a special Wheeler-Balcke forced-draft tower. The results were as follows, the readings given being the average of a five-hour test:

WATER

Gal. per min.	3200
Temp. in	109 deg.
Temp. out	97 deg.

AIR

Temp. in	91 deg., 59 per cent. humidity.
Temp. out	106 deg., 100 per cent. humidity.

The noticeable feature of these readings is the small difference of 3 deg. between the temperatures of the water in and the air out, giving a terminal efficiency of $\frac{106}{109} = 97$ per cent., which is remarkably high for forced-draft work.

Concerning natural-draft towers, but few remarks are necessary. In general, these will require from four to five times the ground area taken up by a forced-draft tower for the same duty, but where space is available they often find favor on account of requiring no power for operation and needing minimum attention. There are two classes of these towers, the open and the inclosed types. The open type is of cheaper construction, without sides, and admits air throughout its entire height. In the inclosed tower the air is admitted around the base only, and in its upward passage it is entirely protected from the cooling effects of the outside atmosphere. Moreover, due to its low velocity, the air leaves the tower at the same temperature as the inlet water; in fact, repeated tests have shown that the terminal efficiency is practically 100 per cent. In addition to this, the sides give a positive chimney effect, which insures a maximum draft and consequently minimum space requirements for this type of apparatus. In the open tower the quantity of air entering at the bottom is necessarily small, due to the absence of draft effect, and its temperature is prevented from approaching very close to that of the inlet water by a continual inflow of outside air along its entire height. As these towers or racks, as they are sometimes called, lack the positive draft created in an inclosed tower, they are necessarily dependent upon prevailing winds, so that, even in the most advantageous spots, their operation is not reliable nor uniform.

As to guarantees, these usually specify the cooling of a certain amount of water through a fixed temperature range under one definite atmospheric condition (usually 75 deg. and 70 per cent. humidity). Prospective buyers often wish a detailed guarantee for some thirty or forty different conditions of temperature and humidity. This the manufacturer is unwilling to make, as it means waiting for payment until all these different weather conditions happen to prevail. In this connection, the curves of Fig. 3 become useful, especially in a natural-draft system, where the amount of air is not given. To illustrate, suppose a tower is guaranteed to cool 1000 gal. of water per minute from 105 to 85 deg., with the usually assumed air conditions of 75 deg. and 70 per cent. humidity, and that it is required to find the cooling under atmospheric conditions of 70 deg. and 50 per cent. humidity. In a natural-draft tower of the inclosed type, with the conditions of test No. 1, the air would leave at 105 deg. and 100 per cent. humidity. By equation (3)

$$H = 8.3\ (1000)\ (105 - 85) = 166{,}000\ B.t.u.$$

By the curves

heat content at 105 *deg. and* 100 *per cent. humidity* = 4850 *B.t.u.*

heat content at 75 *deg. and* 70 *per cent. humidity* = 1800 *B.t.u.*

Heat taken up within the tower = 4850 — 1800 = 3050 B.t.u. Therefore,

$$Q = \frac{166{,}000}{3050}(1000) = 54{,}426 \textit{ cu.ft. per min.}$$

With a given load on the condenser, the heat interchange, the gallons per minute and the cubic feet of air will remain the same; namely, 166,000 B.t.u., 1000 gal. per min. and 54,500 cu.ft. per min. The heat content at 70 deg. and 50 per cent. humidity is 1320 B.t.u. Let X be the heat content of the outgoing air, then:

$$\frac{166{,}000}{X - 1320}(1000) = 54{,}426$$

Therefore,

$$X = 4370\ B.t.u.$$

Locating the point 4370 on the 100 per cent. humidity curve, it will be found to correspond to an air temperature of 100 deg. This is also the temperature of the inlet water under the terminal conditions for this type of tower. By equation (3)

$$166{,}000 = 8.3\ (1000)\ (100 - T);$$

whence, $T_1 = 80$ deg. (outlet-water temperature), representing a total cooling of 20 deg., the same as in the first case. This is obvious when it is considered that both the heat interchange and the gallons per minute remain the same.

By assuming a terminal difference of from 5 to 10 deg., the same method may be applied to a forced-draft system.

For Power Plants at Mines when located near the mine itself an easy and economical method of disposing of the ash is to arrange a tunnel under the fireboxes, with a slope of not less than ¾ in. to the foot. Connect this with a borehole and wash the ashes down the latter with mine water, where they may be used for flushing abandoned workings if desired.—"Coal Age."

Speed Characteristics of Direct-Current Motors

By Alan M. Bennett

SYNOPSIS—The factors affecting the speed of shunt, series and compound motors under different conditions of load and temperature.

Direct-current motors are classified as shunt, series and compound, depending on the method of field winding employed. While this classification is generally well understood, the speed characteristics of these types under various conditions of load, and while attaining their working temperatures, are not so well known. As these characteristics vary greatly in these types, the behavior of the motor under the above conditions becomes an important factor in determining its fitness for certain classes of work.

All motors are supposed to develop their rated speed at full load after having run a sufficient time to reach maximum temperature; and when the speed of a motor is referred to, it is understood to mean that which obtains under these conditions. Variations from the rated speed occur at two periods in the operation, namely, at the time when the motor is started cold and at no load but after the motor has reached its working temperature. The amount by which the speed under the first condition differs from the rated speed is known as the speed variation of the motor; it is sometimes spoken of as the variation from cold to hot. The change from rated speed under the second condition is known as the regulation of the motor. In each case the departure from rated speed is expressed as a percentage of this speed. Both the speed variation and the regulation may differ in motors of the same class and rating, these characteristics depending on design.

In order to determine the changes in speed under the conditions named it will be necessary to note the changes which take place in the factors that influence the speed. It is a characteristic of the electric motor that, under any condition of load and speed, it takes only the amount of current necessary to develop the torque required to do its work at that speed. This it does automatically in the following manner:

Any motor when running, by reason of its conductors cutting flux, must generate a voltage in opposition to that impressed on the motor terminals, and this will tend to limit the current which would otherwise pass through the armature due to its resistance alone. This voltage is termed the counter electromotive force of the motor, and, like that generated in any dynamo-electric machine, is directly proportional to the speed and the flux. For greater convenience in use, the relation shown by these factors may be written

$$e \propto r.p.m. \times \phi \qquad (1)$$

where

e = Counter electromotive force;

ϕ = Flux.

The above expression may be read: Counter electromotive force varies as the speed and the flux. The value of e is always such that the voltage drop in the motor plus e equals the impressed voltage, or letting I represent the armature current, R the resistance of the motor windings and brush contact, and E the impressed voltage; then

$$e + IR = E, \text{ or } e = E - IR$$

Substituting this value of e in equation (1),

$$(E - IR) \propto r.p.m. \times \phi$$

which by transposition gives,

$$r.p.m. \propto \frac{E - IR}{\phi} \qquad (2)$$

This expression, termed the speed equation of the motor, furnishes a basis for determining changes in the motor speed under any condition. It will be seen that the speed varies directly as the impressed voltage minus the IR drop, and inversely as the flux. Therefore, any condition in the motor operation tending to increase either the IR drop or the flux lowers its speed. Likewise, a decrease in either of these factors raises the speed.

Take the case of a shunt motor starting cold and consider the effect of heating on its speed. The voltage on the field of the motor will be the same as that impressed on its terminals, and whether the motor is loaded or not, the field will receive an amount of current depending on its resistance. This establishes the flux of the motor and fixes its speed. As the field rises in temperature its resistance increases and the field current becomes less. The flux passing through the armature is thus lessened, and from equation (2) it will be seen that the effect must be an increase of speed. This condition continues until the field reaches its final temperature. In commercial motors the speed increase from this source will vary from 4 to 8 per cent., depending on the amount of temperature increase and the degree of saturation of the magnetic circuit. The higher the saturation is carried, the less will be the variation in speed.

Temperature increase of the armature affects the speed slightly by reason of increased IR drop. With the motor starting under load there is a certain amount of drop, determined by the motor current and the resistance of the armature. As the armature heats, this resistance increases and with it the drop. From the speed equation it will be seen that an increase in IR drop means a decrease in speed. The amount of variation from this source is always small, however, averaging approximately 0.7 per cent. While for practical purposes it may be neglected, its effect will be seen to be opposite to that caused by field heating; that is, it tends to lessen the motor speed, whereas field heating increases it.

The regulation of the shunt motor, or its change in speed from full load to no load, while not entirely independent of field action, is caused chiefly by IR drop. This drop increases with the load, being the product of the current taken by the motor and the resistance of the armature and brush contact. Equation (2) shows that the effect of increasing the IR drop is to lower the speed of the motor. Therefore, over the range from no load to full load the motor will drop in speed. Regulation, as stated previously, is measured in percentage of full

load speed, and in commercial motors will vary from 4 to 6 per cent.

The effect of the field on regulation is caused by armature reaction. By giving a slight backward shift to the brushes a certain portion of the armature ampere-turns are opposed to the field, which is thus weakened, with the result that the speed is raised. The effect increases with the load and compensates to a certain extent for the decrease in speed caused by the *IR* drop. This shifting, however, can be done only within limits determined by the sparking of the motor.

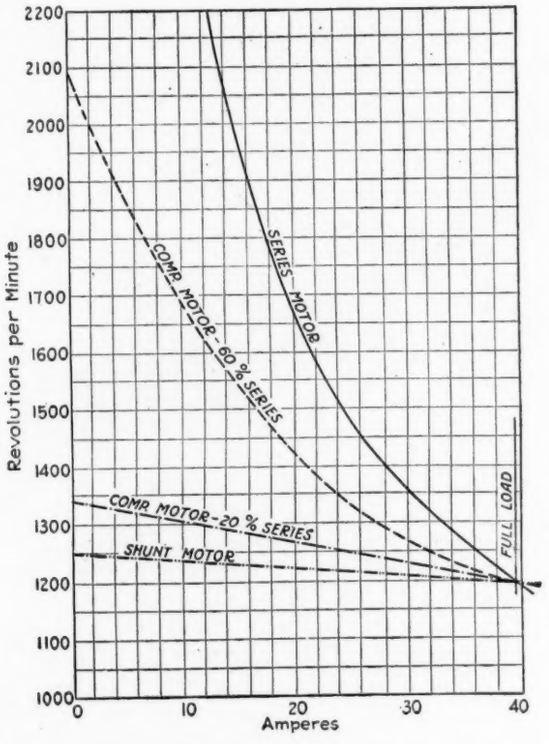


FIG. 1. SPEED CURVES FOR 10-HP. SHUNT, SERIES AND COMPOUND MOTORS

The regulation of motors is most conveniently represented by curves, as in Fig. 1. That for a 10-hp. shunt motor with a 5 per cent. drop in armature winding and brushes shows a decrease in speed from no load to full load of approximately 4.5 per cent.

In the case of the series motor, heating does not play so important a part in its effect on the speed. What variation there is from this source is caused by the *IR* drop only. As the motor heats, the resistance of the armature and series windings increases, and at constant load the drop increases, with the result that the speed is lowered This effect is opposite to that in the shunt motor, where the speed increases with increase of temperature. The amount of this variation in the ordinary series motor will approximate 2 per cent., which is small compared to the change in speed caused by a change in load. In the series motor a change in load affects the speed both through the flux and the *IR* drop. The field current is dependent on the load, being the same as the armature current at all times. The flux passing through the armature being subject to the same variation, the effect on the speed can be seen. Theoretically, the latter would increase from its rated value at full load to infinity at no load. For this reason it is not practical to run series motors without some load in order that they may at all times have a field on them to prevent excessive speed. This is sometimes provided for by winding on a few shunt turns.

The effect of the *IR* drop due to change of load, while greater in the series than in the shunt motor—because there is a drop in the series winding as well as in the armature and brush contact—will still be small compared to the change caused by flux variation. For this reason the speed of the series motor, particularly below the saturation point of the field, will vary almost in exact inverse ratio with the change in flux and, for all practical purposes, the *IR* drop need not be taken into account. The speed curve for a series motor is shown in Fig. 1. For the sake of comparison it is made on the basis of the same armature and strength of field as that of the shunt motor whose curve is shown in the same figure. The rapid increase in speed as the load decreases will be noted.

The compound motor, having a field composed of shunt and series windings, partakes somewhat of the characteristics of both the shunt and the series motors. The speed will increase with the heating of the shunt field, as in the case of the shunt-wound machine. Heating of the armature and series field will cause an increased drop in those parts, with a consequent lowering of the speed. However, this drop and the resulting change in speed will be greater than in the shunt motor on account of the added resistance of the series field, the amount depending on the percentage of series windings carried by the motor.

Regulation will depend both on the *IR* drop and the change in flux due to the action of the series field. Of these two factors, the latter will have the greater effect. In fact, in those cases where the series field strength is in excess of 20 per cent. of the total, the change in flux will practically determine the regulation. As the percentage is increased, the change in speed with the change of load becomes more pronounced and approaches more

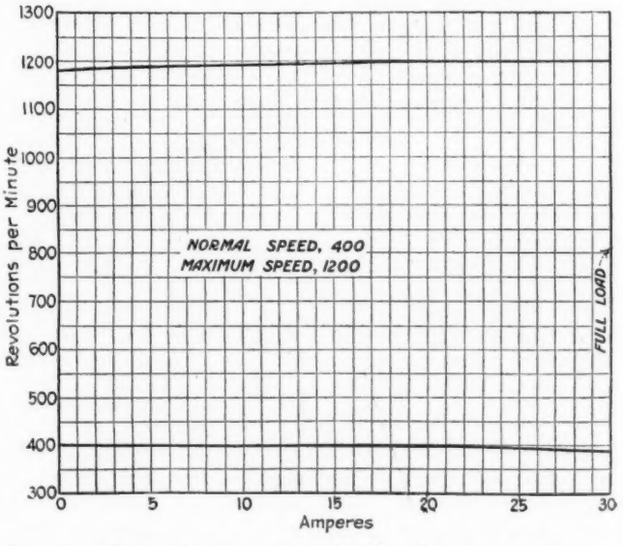


FIG. 2. SPEED CURVES FOR 7½-HP. INTERPOLE SHUNT MOTOR

nearly that of the all-series motor. There is the difference, however, that, due to the shunt field, the compound motor is not deprived of all its flux at no load, and there is not the danger of excessive speed, as in the case of the series motor. In Fig. 1 are shown two speed curves of compound motors, one having 20 per cent. series field,

and the other 60 per cent. In the former the regulation is as close as 12 per cent., while in the latter it approaches very nearly that of the series motor. Both of these curves are on the same basis as regards armature and total field strength, as the curve for the shunt motor.

The compound motor, as generally known, has its series field so connected as to strengthen the effect of the shunt field as the load increases, and thus to gain some of the benefits obtained with the series motor, namely, a powerful starting torque and rapid acceleration. By connecting the series winding so that it opposes the shunt field, there is had what is known as the differential motor. This is done to compensate for IR drop from no load to full load and render the motor constant in speed over that range. As the load increases, the flux established by the shunt field is lessened by the action of the series, with the result that the speed, instead of falling off due to the IR drop, is maintained constant. The differential motor is not widely used, however, on account of other features which are a disadvantage, among these being low starting torque and lack of ability to stand overloads.

In the three regular types of motors described, it has been seen that the IR drop causes a falling off in speed from no load to full load, this being the least in the case of the shunt motor. Also, up to certain limits this can be compensated for by shifting the brushes and getting the effect of armature reaction on the field. In a motor fitted with interpoles this same compensation will be noted, only in a more pronounced degree. The action of the interpoles has a weakening effect on the main field similar to that produced by the armature. The result is an increase in speed as the load comes on. This action may even be so exaggerated as to cause a higher speed at full load than at no load, particularly with adjustable-speed motors at their weak field points. There is the difference, however, in the case of the interpole motor that the brushes must be kept at the neutral point on the commutator and brush shifting cannot be taken advantage of.

Fig. 2 shows speed curves for a 7½-hp. interpole shunt motor having a normal speed of 400 r.p.m., and a maximum of 1200 r.p.m. by means of field control. At the low speed it will be noted that the regulation is within 1½ per cent., the speed being practically constant, while with a weak field there is an actual increase of speed from no load to full load.

Powell Automatic and Double-Automatic Stop Valves

The Powell double-automatic stop valve prevents steam from entering a boiler not under pressure from a header, even should a handwheel be operated for this purpose, as the valve cannot be opened by hand until pressure is again raised in the boiler, but it can be closed and screwed down tight by hand. It also prevents steam from leaving the boiler in case of an accident to the pipe line.

The valve disk is of one piece, top and bottom guided at A and B, Fig. 1, and has a seat on both sides at C and D. The disk seats at C should anything happen on the boiler end and on D if there is any trouble on the main steam line.

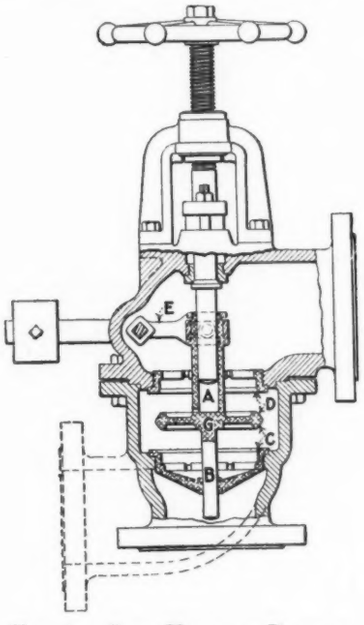


Fig. 1. Side View of Double Stop Valve

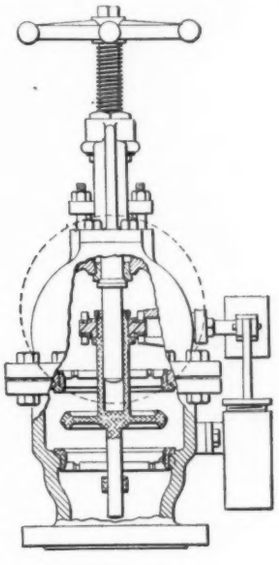


Fig. 2. End View of Double Stop Valve

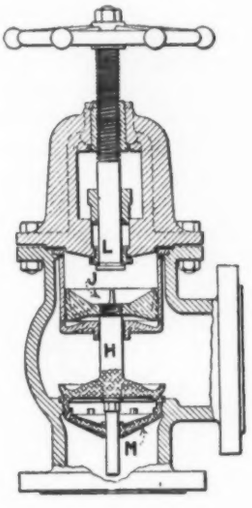


Fig. 3. Section through Single Disk Valve

The upper and lower valve seats are of special nickel composition to resist the corrosive action of the steam. The fork E is fastened to the balancing lever stem and holds the valve disk G in position, the weight balancing the valve disk. The oil dashpot, Fig. 2, the chamber of which is filled with oil, is to prevent the valve from chattering.

The valve also acts as an automatic equalizing valve between a battery of boilers and can be used as an ordinary stop valve by screwing down the stem onto the disk.

In the automatic stop valve, Fig. 3, the valve disk H is attached to the plunger J. The disk and plunger are made with but one screw part and are permanently fastened with two sets of screws, making practically one solid piece, guided at both the upper and the lower end. The dashpot fits snugly in the valve body. The rim of the upper part of the disk plunger J is grooved to work with a minimum of friction and respond readily to any varia-

tion in the pressure. The dashpot has two vent holes at the top and two at the bottom, to allow draining any condensed water that may collect therein. As the lift of the disk is equal to the depth of the dashpot, a full opening of the valve is insured. The height of lift is regulated as desired by raising or lowering the screw stem *L*.

The seat ring *M* is made with a guide for the lower part of the valve disk. When necessary, the seat can be readily renewed by inserting a flat tool between the lugs projecting from the circle and unscrewing. This valve will automatically shut off the flow of steam from the header to the boiler in case a tube should burst. It can only be opened by the pressure in the boiler, thus acting as an automatic equalizing valve between the battery of boilers, and also making it impossible to accidentally turn steam into a boiler when it is being cleaned.

These valves are manufactured by the William Powell Co., Cincinnati, Ohio.

Condensation in Hot-Blast Heaters

By James D. White

In the design of a hot-blast heating system it is necessary to know the number of pounds of steam required per minute to heat the specified volume of air through the desired range in temperature. Multiplying the cubic feet of air per minute by the weight of one cubic foot by the specific heat of air and by the temperature rise gives the number of British thermal units required. Dividing this product by the latent heat of steam at the average pressure in the heating coils gives the number of pounds of steam required per minute. Expressing this as a formula:

$$C = \frac{C.F.M. \times W \times S \times T}{L}$$

In which

C = Condensation in pounds of steam per minute;
$C.F.M.$ = Cubic feet of air per minute passing through the heater;
W = Weight of one cubic foot of air in pounds;
S = Specific heat of air;
T = Rise in temperature through the heater;
L = Latent heat of steam.

If steam is purchased from an outside source the formula may be used to determine the service to be provided. If steam is generated in the building it will determine the boiler-horsepower required, or the pounds of exhaust steam from the engines or turbines.

The main supply pipe to hot-blast heaters is often made too small, due to the fact that the actual requirements are not carefully considered. A rule of thumb common among steam fitters is to make the supply pipe equal in size to that required for direct radiation of five times the surface contained in the hot-blast heater. Cast-iron direct radiation for low-pressure steam will transmit about 250 B.t.u. per sq.ft. per hr. On the rule of thumb basis this would mean a transmission of 1250 B.t.u. per sq.ft. per hr. for the hot-blast heater. This transmission rate is correct for deep heaters with low velocities over the surface. For shallow heaters or high velocities this rate may run as high as 2500 B.t.u. per sq.ft. per hr., showing the error that may be made by the use of the above rule.

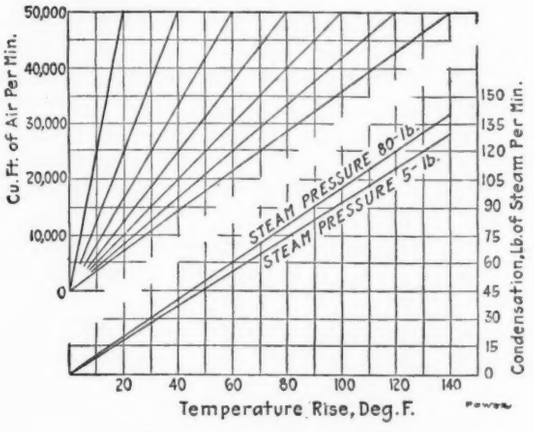
The following values are taken from published data regarding a well known type of hot-blast heater:

Number of Sections	Temperature Rise
1	32
2	59
3	82
4	100
5	116
6	129

From an inspection of the above values it will be noted that nearly one-fourth of the total heat rise is obtained in the first section of the heater. Since condensation is directly proportional to heat rise it follows that nearly one-fourth of the total steam required by a heater will be condensed in the first section.

It is customary to tap all sections of a hot-blast heater for the same size of pipe, this size being ample for the first section of the heater. There appears to be no good reason for this since a substantial saving could be made by tapping the sections in proportion to their steam requirements. The amount of condensation as calculated above is also useful in determining the size of steam trap or pump for returning the water of condensation back to the boiler.

The standard curves of hot-blast heater manufacturers are usually based on the heat rise with velocities of air



Condensation Curves

measured at a standard temperature of 70 deg. This temperature of air has been assumed for the volumes of air as given in the condensation curves presented herewith.

As an example, assume that it is desired to know the pounds of steam required to heat 40,000 cu.ft. of air per minute from zero to 100 deg. with steam at 5 lb. pressure. Starting at the line in the upper left-hand corner of the diagram corresponding to 40,000, draw a horizontal line until it intersects the 100-deg. line. Drop a vertical line intersecting the steam-pressure curve; then draw a horizontal to the right and find that the condensation is 74¼ lb. of steam per minute.

Safety Suggestions—If any employee shows signs of carelessness, tell him about it. If he persists, report him. Otherwise you may be the one to receive an injury through his negligence. Careless men are a menace to all around the power station.

Don't forget the other fellow. Careful men are hurt and killed every hour through the negligence of others. Better be safe than sorry.

Don't fail to look up occasionally; there is danger from above as well as below.

Don't fail to have any defective tools or equipment repaired at once.

Always treat conductors and switches as though they were alive.—"Aera."

Editorials

Obstructive Legislation

Large power interests, hiding behind the plea of states' rights, are attempting to defeat the Ferris water-power bill by creating a situation that will prevent the measure coming to a vote in the Senate before March 4, and there is growing apprehension that the same interests which have prevented conservation legislation for the last ten years will still be able this year to keep the natural resources in the public domain out of use and development.

It is a big game for big stakes that large financial and political interests are playing. Hydro-electric power is still in its infancy. Its possibilities are hardly to be conjectured. Even under present conditions of development, there is enough potential water power in the United States to take the place of the total coal consumption. In some parts of the country and under some conditions of construction, production of power from coal entails a lower installation cost than hydro-electric production, even where water power is available. Constantly improving methods of hydro-electric development and improving methods of power transmission, however, make it impossible for anyone to say what this water power will be worth or what its development will cost in the next decade or two. When the diminishing supply and increasing price of coal are considered, water power assumes an importance that can hardly be over-estimated.

So far-reaching were the plans and so rapid has been the progress of the hydro-electric monopoly that in 1913 twenty companies or groups of financial interests, all more or less closely interrelated, had acquired control of 2,710,866 of the 7,000,000 horsepower developed in the United States, and these same combinations also controlled 3,556,500 undeveloped horsepower. In California one corporation owns 27 per cent. of the total developed horsepower in the state, and two groups own 57 per cent. of the total development; in Oregon 90 per cent. of the developed horsepower is controlled by these allied groups.

With a few exceptions, the power sites not remaining in the ownership of the Federal Government have passed into private ownership in perpetuity. State governments generally have been notoriously profligate in giving away public property and franchises for small return or for no return, and until recent years, generally with no provision for regulation. In many sections state governments, to induce development and industries, have made it a rule not only to give away such valuable public assets, without price or restriction, but also to exempt such grants or gifts from taxation for a longer or shorter time.

Conservative estimates place the total available water power in the United States at 25,000,000 horsepower, of which 7,000,000 horsepower has been developed. Of the undeveloped water powers, about three-quarters are owned by "Uncle Sam," the power sites being located on public lands. Control and monopoly of this undeveloped power are the stake for which the big interests are playing.

To prevent more of the remaining water powers being grabbed by private monopoly through fraud or misrepresentation, the Government several years ago located all the valuable power sites in the public domain, and the President withdrew them from entry. Most of them are still withdrawn. Special interests protested, but the prevailing public sentiment was in favor of conservation, and then the interests changed their tune. Now they are all for states' rights. They are for conservation, and they are for regulation, but they want the Government to give the lands to the states and let the states do the regulating. The reason is that they can do business more profitably with state governments, governors, legislatures and utility commissioners than they can with the President, Cabinet officers, Congress and the Interstate Commerce Commission.

Secretary Lane, who is a Western man and who wants to see the resources of the West developed, has been backing legislation to encourage the use of water power and the mining of coal, oil, phosphates and potash under a leasing system. President Wilson supports this policy. The House of Representatives has passed the Adamson and the Ferris water-power bills, drawn along these lines. The Ferris bill provides that power sites may be leased for a term of not more than fifty years, with a twenty-year renewal of the lease at the end of the fifty-year period, on terms to be fixed by the Secretary of the Interior. At any time after the end of fifty years, the Government would have the right to take over the power plants by paying their fair value to the owners, and thereafter to either operate the plants as government institutions, or to sell or lease them to municipalities, states or individuals.

In support of this legislation, the President, Secretary Lane, such conservationists as Gifford Pinchot, and others have urged that a fifty-year permit would enable power companies to enlist capital and finance their operations profitably, giving the communities the advantage of early development and insuring against monopoly and extortion, with a reversion of these valuable rights to the public at the end of the franchise term.

Some of the big power interests have agreed to this program; others are opposing it, some of them openly, but more of them under cover. Senators who for years have been notoriously representative of big interests and special privileges are opposing the bill. The opposition, however, is subtle. It is not openly in opposition to limited franchises or regulation. To defeat the legislation, the old Civil War issue of states' rights has been dug up. Corporation lawyers, bankers, politicians and lobbyists are busy telling the Senate that the public lands, including the mineral resources and water powers, should be given to the states in which they are located.

The corporations and the politicians know what the states have done in the past with their resources and what they may reasonably be expected to do in the future. State utility commissions generally have few powers, and do not exercise them. Some of the Western states have no utility commissions and have never made any pretense of regulating anything. Mineral lands, water powers and similar natural resources belonging to these states have

often been considered fair loot for anybody who could get them.

There are votes enough in the Senate to pass the bill if it can be got to a vote, but the opponents are prepared to play all the legislative tricks they know to prevent this. If they can stave off a vote until after March 4, the bill will then be dead, and a new bill will have to be passed by the House, which means more years of delay and disuse for the water power.

Purchasing Coal

The question of the ultimate advantage of purchasing coal on the B.t.u. basis is raised by Mr. Brownell on page 131 of this issue.

It is true that all of the coal mined must, or should, be used as fuel, and if the dealer cannot market inferior stuff in one place he will send it to some other place. This is, of course, an unavoidable condition of affairs to the dealer, and may be entirely satisfactory to the user so long as he is equipped to handle such fuel economically; but the purchaser or user should know what he is paying for or trying to make steam with. The crux of the whole matter is as the pure-food advocate, Dr. Wiley, has often said—he has no desire to dictate to or to prevent any individual from using adulterated foodstuff, but he does want him to have every opportunity of knowing what he is using and what the probable effect will be. Likewise in the use of coal, the consumer should know as much as possible of the characteristics of the available fuel supply in order to modify his furnace and equipment to suit the conditions. The total cost in fuel, labor and upkeep of evaporating water is the basis of comparison and the final test of the value of fuels. If the consumer has no alternative he must take what he can get and make the best of it.

13.8 Kw. per Boiler Horsepower

At a recent lecture by W. A. Blonck, of Chicago, before the New York Electrical Society, the interesting information was given out by Mr. Pigott, of the Interborough Rapid Transit Co., that eight boilers of 520 horsepower each are to serve 30,000-kilowatt turbo-generator capacity. This gives 7.21 kilowatts per boiler horsepower, which is considerably higher than that heretofore practiced. The Connors Creek station of the Detroit Edison Co. was designed so that in emergencies 5.65 kilowatts per boiler horsepower might be attained. This was the highest ratio heretofore.

The statement that so and so many kilowatts of output is obtained from a boiler horsepower does not tell much about the rating at which the boiler must be run unless the water rate of the prime mover is given, because the ratio is dependent both upon boiler rating or capacity and the steam consumption of the unit or units served by the boilers. The ratio is, however, an indication of the advance in boiler practice and turbine economy.

In this respect the Interborough's Seventy-fourth Street plant is interesting. The steam consumption of the 30,000-kilowatt turbine, served by eight 520-horsepower boilers, is 11.25 pounds at normal load. This gives 3.06 kilowatts per boiler horsepower at the normal rating of the boilers. So at a little over 200 per cent. of rating these eight boilers carry the load of 30,000 kilowatts, giving 7.21 kilowatts per boiler horsepower. A statement to us by Mr. Stott, superintendent of motive power for the Interborough Rapid Transit Co., that there might be times when it would be necessary to operate the boilers at 450 per cent. of rating for short periods, as in emergencies, is of much interest here. A boiler rating of 450 per cent. is possible, for with a clean and well designed boiler and furnace, the only limit to capacity is the amount of fuel that can be burned on the grate. As 3.06 kilowatts per boiler horsepower is obtained at 200 per cent. of rating, at 450 per cent. 13.8 kilowatts could be had per boiler horsepower.

James Watt must turn over in his grave at this.

Summary of Boiler Accidents

The list of boiler explosions which occurred during the first half of the year 1914 contains 320 as the total number. Of the number mentioned in our Jan. 5 issue, 20 are not included, because of denials on further inquiry that explosions had occurred. It frequently happens that in reply to our inquiry to the parties concerned a complete denial is received, but later information verifies the original report. This, in some cases, may have been due to an interpretation of the term "explosion" as a violent disruption of the body of the vessel used to generate steam. In general, for the purpose of tabulation the word is taken to apply to any failure which even temporarily puts the boiler out of use, including tube, header and blowoff-pipe failures. The nature of the failure is stated in every case in which the facts are obtainable. These statements are not always as full and satisfactory as might be desired.

The greatest number of accidents from any one cause was due to tube failures, but cast-iron header failures show an alarmingly increased percentage. When it is considered that from four to ten or more tubes are connected to a pair of headers and that a number of types of water-tube boilers in use are not so constructed, the total header failures compared with tubes is enormous.

Next in number comes the blowoff pipe. Considering the severe service and exposure of these pipes it is not surprising that they should deteriorate rapidly. This being generally recognized, it is evident that this part of the boiler should receive more careful and frequent scrutiny and should be replaced on the first appearance of weakness or danger. Cast-iron heating or domestic boilers are shown to be frequently neglected and mismanaged. In view of the damage done (in some cases well up in the thousands) when one of these boilers explodes, it cannot be said that they are receiving the inspection and supervision they should have. Even kitchen ranges and boilers have contributed a considerable amount to the total wreckage. A comparison of the totals for the first half of 1913 with those for the same period in 1914 follows:

Total number of accidents, 264 (1913), against 320 (1914), with a loss of life of 53 against 120, and injured, 192 against 240. The monetary loss was $193,000 (1913) and $246,000 (1914), respectively, with an average for those for which estimates were obtainable of approximately $1330 against $3000. Tube failures appear to have been more numerous during the year 1913, as 70 are shown, while a year later only 60 are reported. Header failures were only an incident, however, in 1913, while 30 occurred in 1914. Blowoff accidents stand 13 to 17, and cast-iron heating boilers 33 to 70 for the two periods, respectively.

Correspondence

Purchasing Coal

The writer recently had occasion to condemn a car of No. 1 buckwheat coal on account of excessive slate and screenings. A representative of the coal company arrived a few days later, and upon screening a 50-lb. sample from the car he found 12 per cent. rice or barley coal. A 10-lb. sample showed 14 per cent. slate.

The inspector passed the car, as he said the percentage was well within the allowance. Upon being pressed for some definite understanding as to what we were forced to accept, he made the statement that the coal company allowed itself 15 per cent. screenings and 15 per cent. slate. He was then asked if his company sold anthracite on a heat-unit basis. He replied that it would not sell coal on such a contract. Consequently, if the slate test is the only one which is acceptable, it becomes the test we are forced to use.

The large consumer situated on a navigable stream or where he can be served by more than one railroad can buy his coal on the heat basis, and he has this advantage over the smaller consumer who can obtain coal from only one railroad.

After all, does it pay? Is it to the consumer's advantage? Buying coal on a B.t.u. basis resolves itself into a matter of service. The consumer pays for that service whether he, the coal company or some disinterested party analyzes the coal, and when the cost of this service is put over against the gain, will not the apparent saving effected be wiped out?

R. A. Brownell.

Middletown, N. Y

Concrete for Furnace Lining

A concrete that could be used successfully as a boiler-furnace lining would do much to reduce the maintenance cost of the furnace. The largest item in furnace maintenance is labor, and when the labor is inexperienced, as when a bricklayer who never laid firebrick is engaged for the job and insists on laying the brick with a $\frac{3}{16}$-in. joint, the cost of upkeep is indeed high.

A concrete containing limestone would be unfit for the purpose because limestone calcines at a comparatively low temperature. The writer's experience with concrete furnace lining is limited to a test in which two patches 18 in. square were made, one on each side of the furnace, one patch being of slag concrete and the other of cinders. Neither proved satisfactory, probably because the concrete was not given sufficient time to set, the boilers being put back in service within 36 hours.

A concrete that may fulfill the requirements could be made of portland-cement clinker, graded from fine to coarse so as to make unnecessary the addition of sand. This concrete should be made with a minimum of water. Such material is sometimes used as a lining for cement kilns. Engineers in cement plants may be able to give some information on the use of clinker concrete in boiler furnaces.

Could boiler settings be made of concrete a great convenience would be effected; the two-inch air space is no longer fashionable, and the form work being simplified thereby, all the work could be done by the boiler-room force, thus eliminating the bricklayer.

C. O. Sandstrom.

Kansas City, Mo.

I have tried several different mixtures of cement and sand with hard-coal ash, soft-coal ash, flue cinder and salt for furnace lining, bridge-wall and fire-door arches. A mixture of one part cement, three parts hard-coal ash, one part fireclay, one-half as much sand as cement and about one per cent. salt made a bridge-wall that outlasted firebrick. For lining, I have been able to get a cement mixture that would outlast a good firebrick lining when properly laid, if there is time to let the wall harden thoroughly before it is necessary to start the fire.

The mixture described in Power, Dec. 15, p. 840—i.e., one of cement to five of hard-coal ash and one-half of sand—ought to be good, but I would add about one per cent. of salt to the mixture. When the wall gets hot the salt and sand will tend to melt and will fill up the pores and cracks showing in the cement.

For fire-door arches about the same conditions prevail. One part of cement, one of sand and five of hard-coal ash or broken soft-coal cinders or clinkers holds up better than firebrick set with fireclay. It is necessary, however, to let the arch have at least four weeks to set before putting the furnace into operation, and in many cases this is not possible.

A. A. Blanchard.

Oxford, N. J.

Graphite in Boilers

Much has been said in Power relative to the use of graphite in boilers. I first used graphite in a plant having four 300-hp. water-tube boilers. No compounds had been used in this plant and the tubes were in fair condition. It required, however, from twelve to fifteen minutes to get through a tube with a turbine cleaner, maintaining 180 lb. water pressure at the turbine, and occasionally we encountered tubes which required twenty to thirty minutes, but fifteen was a fair average. Before using graphite we turbined the boilers every ninety days, and this practice was followed for about nine months after it had been in use.

Notwithstanding that the output of the plant was increased more than 30 per cent. during the first nine months' use of graphite, we were enabled to increase the continuous runs of the boilers. Records show that two boilers were operated for 139 and 143 days, respectively, without turbining any of the tubes. Then they were opened and turbined throughout without difficulty. Furthermore, we were able to get through most of the tubes in less than eight minutes for each. Graphite did not show favorable signs until it had been in use more than

five months and did not get in any good work until we had used it for about seven months; at this time one and one-half barrels had been used.

At the end of seven months we began loosening large pieces of scale in the steam drums. The writer put two good men in the drums for over two days, who succeeded in getting off large quantities of scale which ranged from $\frac{1}{8}$ to $\frac{5}{16}$ in. thick.

In my next plant I found dirty boilers and lost no time in ordering a turbine cleaner and a barrel of graphite. It required more than five months to get the cleaner, and this gave the graphite a chance to act before its arrival. I find that graphite will soften incrustation and loosen the heavy scale in the drums or on the sheets; it is, however, essential that mechanical methods be employed to remove the scale.

If maximum results are to be realized from the use of graphite, the boilers must be cooled down thoroughly before they are opened, and the drums must be washed immediately with a large hose and high water pressure. The scale will be soft mud when wet, but it will get hard when it dries.

It is much easier to wash the surfaces and then scrape them than to allow the accumulation to solidify and then pound it loose with the peen of a hammer. Do not take the tube caps or the man-heads off and allow a boiler to stand over night before washing and turbining. Take the tube caps off and put the turbine through the tubes as fast as possible. Then if any scale of consequence still remains, replace the cutters on the turbine cleaner with new ones and go through the tubes carefully the next day.

WALDO WEAVER.

Emporia, Kan.

Underfeed Stokers

In Osborn Monnett's article, "Underfeed Stokers," in the Dec. 15 issue of POWER, there is illustrated a "typical setting of American Stoker" under a return-tubular boiler. The stoker shown is not the American Stoker, but is the Type "D" of the Combustion Engineering Corporation. This stoker was formerly made by the American Stoker Co., and the drawing from which the illustration was made is of an installation made by that company. Mr. Monnett's assumption was natural, but it is regretted that he did not have the data on the Type "E" stoker for his article.

The Type "D" stoker is designed primarily for internally fired boilers with cylindrical or corrugated furnaces, but it may be applied to other boilers of 100- and 150-hp. capacity.

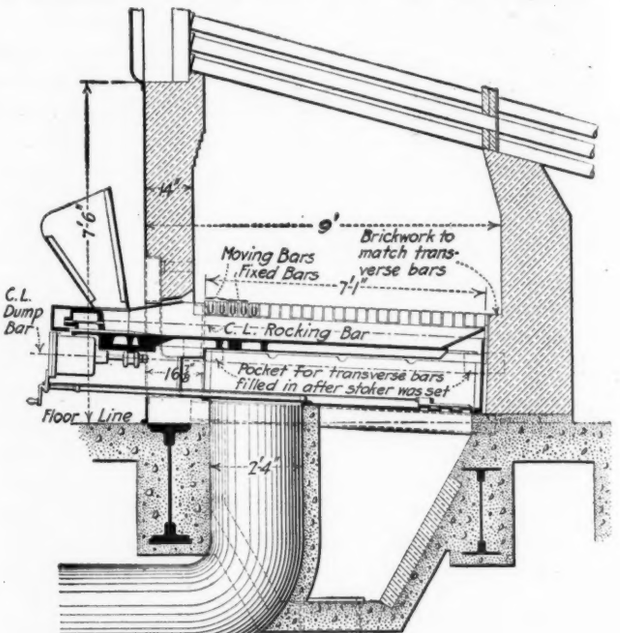
For boilers of 200 hp. and upward, the Type "E" stoker of the Combustion Engineering Corporation will effectually prevent smoke at ratings of 150 and 200 per cent. But one retort is installed in furnaces up to 12½ ft. wide.

The illustration shows a typical setting under a horizontal water-tube boiler. For vertically baffled boilers the minimum height of setting is 7 ft. 6 in. from the floor line to the header, and on Eastern coal of 16 to 26 per cent. volatile matter, 50 lb. per sq.ft. of furnace area may be burned without smoke. This is equivalent to 175 to 225 per cent. of the boiler rating. For Western coals of 30 to 40 per cent. volatile matter the height of the setting should be 8 ft. or more for similar results. There will, of course, be proportionately larger grate surface for Western coal if high ratings are required.

OPERATING

When starting fires with a Type "E" stoker, fill the retort and over-grates with coal about two inches deep, throw several shovelfuls of live coal along the retort and on the grates and start the blower slowly, increasing the speed as the coal becomes ignited. As soon as the coal is well ignited, start the stoker, increasing the coal feed and the air supply as required.

The fires should not be carried at over 8 to 12 in. in thickness with coking coal, or 4 to 8 in. with free-burning coal. The distribution of coal is uniform, making it unnecessary to poke, slice or rake the fires. The automatic regulation will take care of any variations in load between 100 and 200 per cent. of rating.



TYPE "E" STOKER UNDER A WATER-TUBE BOILER

When ashes have accumulated on the dump trays to such a depth that the trays will not hold more, they should be dumped. Slow down the stoker slightly and burn out all coke on the trays, using forced draft if necessary. Drop the dump trays and if any clinker has adhered to the side walls or overhangs the ends of the fire bars it can easily be removed with a slice bar. Then restore the trays to the running position and operate the stoker a little faster than normal for a few moments; then throw in the automatic. Ashes should be dumped every two to four hours under ordinary conditions. The ash on the fire bars should not be disturbed.

When banking fires, shut off the air and feed in enough coal to make the required bank, then close the fire-doors and damper. If necessary the bank may be replenished by feeding in more coal.

To start from a banked fire, dump the ash, open the air gate a very little for a minute, break any large lumps of coke and then start the stoker, increasing the coal feed and air pressure as required. To kill the fire in case of accident, shut off the air, drop the dumps and feed coal

to maximum capacity. Turn a hose into the ashpit to quench the live coal dumped.

JOHN VAN BRUNT,
Combustion Engineering Corporation.
New York City.

Centrifugal Pumps for Boiler-Feed Service

We have read with interest the extract from a paper by E. S. Adams, in the Dec. 29 issue, p. 934. From our experience we are inclined to disagree with this statement which appears on page 935:

While this is especially true with motor-driven pumps, it also applies to turbine-driven pumps, as a serious overload (due to the enormously increased capacities which are obtained on some designs of pumps when the pressure is dropped) sometimes destroys the turbine.

The condition mentioned by Mr. Adams is impossible, as an overload beyond the designed capacity of the turbine would only cause it to slow down. We would be interested to learn of any turbine-driven installations where the action referred to has apparently taken place.

RAINES KESSLER,
Terry Steam Turbine Co.
Hartford, Conn.

Thick Boiler Plates

The article by S. F. Jeter in the issue of Dec. 22, p. 884, recalls to the writer a battery of horizontal tubular boilers in which the plates are $\frac{23}{32}$ in. thick (just $\frac{1}{32}$ under $\frac{3}{4}$ in.). At the girth seams the plates are reduced to $\frac{7}{16}$ in. in a way similar to that shown in Fig. 2 in Mr. Jeter's article, except that in this case, and in fact in all cases of heavy fire sheets that have come to my notice, both the inside and the outside plates have been reduced in the manner shown in Fig. 1 herewith. These boilers have been in service day and night for 12 years. Stokers are used and a steam pressure of 145 lb. is carried. The

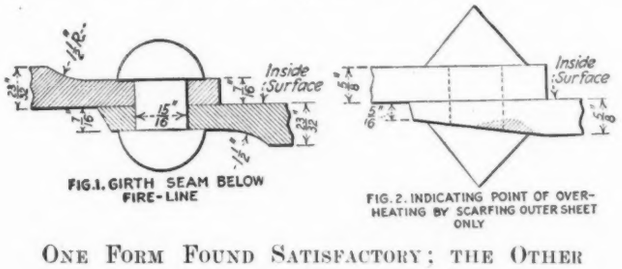


FIG. 1. GIRTH SEAM BELOW FIRE-LINE

FIG. 2. INDICATING POINT OF OVERHEATING BY SCARFING OUTER SHEET ONLY

ONE FORM FOUND SATISFACTORY; THE OTHER QUESTIONED

service required of the boilers is much more severe than in the average plant, yet there has never been a leak or a fire crack at the girth seams.

Plates $\frac{23}{32}$ in. thick are the heaviest I have known in an externally fired boiler, and I have seen them with $\frac{9}{16}$-in. plates with the thickness of each plate reduced at the girth seams to about half the original thickness and none of them ever developed defects at this seam.

As Mr. Jeter points out in his Fig. 4, there is a tendency to crystallize at the calking edge when exposed to high temperatures. Crystallization of the plate at the point indicated might result in a fire crack extending from the edge of the outside sheet to the rivet hole, not a very serious defect and one easily repaired. In his Fig. 5 he suggests a method which would no doubt overcome the tendency to overheat at the edge of the plate, but which would transfer the crystallization to a point farther back, or about as shown in Fig. 2 herewith. A fire crack developing at this point would open into the solid plate.

Experiment or practice has not yet determined the heaviest plate practicable in an externally fired boiler. We do not know, however, that fire sheets ½ in. thick stand up well under heavy firing with two full thicknesses of plate at the girth seam and that plates ⅝ in. thick will fire-crack with a double thickness of plate under comparatively easy conditions. The limit of thickness at the girth seam may or may not lie within this range, but we do know that the girth seam shown in Fig. 1 has stood up well under severe conditions for a number of years, and the thickness of the joint is less than that made by two ½-in. plates, and that the tendency to crystallize at the calked edge due to high temperatures is less than it would be in a girth seam with a total thickness of 1 in. through the joint.

THOMAS GRIMES.
Houghs Neck, Mass.

Cost of Steam

The cost of steam regardless of its use is an important matter to every live engineer. The information on this subject which occasionally appears is interesting and valuable, as it shows us what the other fellow is doing with his equipment, fuel, etc., and enables us to compare notes. I think we can stand a generous amount of steam cost data, but to be of maximum value the reports should be fully and accurately comparable. Most of them simply show that A has a small plant and pays $4 a ton for coal, and give a figure on the cost of steam per thousand pounds, based on the cost of labor and coal only.

B has a different plant, equipped with all of the latest devices for producing steam cheaply, and is able to buy and use a low-grade coal which gives an equivalent heat value. He therefore produces steam for much less money than A, which makes A feel like thirty cents when he sees it. Of course, we care nothing about A's feelings. Besides, it may wake him up and do him good. There is doubt, however, whether B has a right to feel so chesty. We have but part of the actual cost to A and perhaps less information from B, therefore, a fair comparison is out of the question.

It may be satisfactory enough to charge only coal and labor as the cost of steam as a means of comparing different coals, or for other comparisons within that plant, but when an attempt is made to get at the actual cost of steam or compare one plant's results with those of another, the need of reducing the results to a common basis is evident. This means a full description of the equipment, repairs, interest on the investment, insurance, taxes and depreciation. The cost of operating draft machinery, stokers, etc., should be included with the cost of labor and fuel. Until this is done no one can discuss relative steam costs intelligently. Any useful information is acceptable, but full and exact information is needed.

H. L. STRONG.
Yarmouthville, Maine.

Shaft Coupling Made into a Belt Pulley

While employed at Fort Yuma Indian School, I needed a six-inch belt pulley for a wood saw. As some readers may know, getting such supplies on an Indian reservation is a long and trying ordeal. Therefore, I used a flange coupling keyed to the saw mandrel, with thick planks bolted between and sawed to the same circle as the flanges. This served the purpose while we sawed 100 cords of wood for our winter's supply.

J. E. Strother.

Rock Island, Ill.

Calculating How to Cut a Manhole Gasket

The following questions were asked me recently in a steam-engineer's license examination. I also give my answers:

Could you cut a manhole gasket if the manhole head was not convenient to be used and do it correctly?

Yes, if I knew the distance between the two foci of the ellipse.

Given the long and the short diameters, how would you find the distance between the two foci?

Square half the long diameter, from it subtract the square of half the short diameter; the square root of the remainder is half the distance between the two foci.

I spent three days hunting for the answer to the last question, and I found another item that helps to show why the above answer is true.

To locate the two foci of an ellipse graphically, take one end of the short diameter as a center and half the long diameter as a radius and describe arcs intersecting the long diameter each side of the short diameter. These points of intersection are the two foci.

F. C. Wires.

Seattle, Wash.

New vs. Second-Hand Machinery

M. E. Griffin's experience (Power, Dec. 22, p. 889) with the man who wanted an exceptionally economical engine and finally purchased an old second-hand one reminds me of some of my own experiences with would-be cheap-power-plant owners.

A man in southern Florida hired me to install a three-ton ice machine and set up a new (?) sawmill. When I arrived I found an old belted compressor rated at three tons, but which its former owners had never been able to drive past one ton of ice in 24 hours, with a one-ton capacity brine tank and a cold-storage room taking approximately one ton of refrigeration for 24 hours, an old internally fired boiler all pitted and having several soft patches on it, also an old locomotive-type boiler with an engine mounted on it.

I set up the outfit, but refused to run it, and the last I heard it was still idle because it is cheaper to buy ice and pay freight on it than to make it with the outfit. The sawmill part of the equipment was a little better, and after getting a new boiler the owner finally did get a man to run it.

I was engaged another time as a millwright and master mechanic for a lumber company, the president and superintendent of which were experienced millwrights. The boiler and engine were too small, so they made up their minds to install larger ones. I advised them to get a new boiler and engine and get them from a reliable manufacturer, but they decided this was too expensive and finally bought a second-hand Scotch boiler that had been run about six months and then replaced by a return-tubular boiler of the same rated capacity, because the Scotch boiler would not do the work required.

I begged both the president and the superintendent not to take the boiler, as it was entirely unsuited to their work and to the fuel that they must use, and pointed out other defects, but they got it and the results turned out as I said; the furnace was too small to burn the fuel, and after trying several kinds of grates they finally inclosed the boiler in a brick setting, making virtually a double furnace. The outfit cost $250 more when finished than a new return-tubular boiler with a full brick setting would have cost.

The engine they bought was an old sawmill engine which had been over twenty years in service and through two fires. They paid $150 for it and $300 to have it repaired. A better engine could have been purchased new for less than $500, and would have been an engine, not a junk heap.

Why men will pay good money for worthless piles of scrap iron in the shape of second-hand engines and boilers I could never understand, for new outfits are always cheaper in the end.

A. A. Blanchard.

Oxford, N. J.

Testing Out Automatic Safety Devices

The editorial on "Testing Out Automatic Safety Devices," in the Dec. 1 issue, reminded me of a plant in which I was oiler. The engines were cross-compound Corliss, with the governor on the low-pressure side and a safety stop valve in the high-pressure steam pipe, operated by fly balls on the high-pressure side. These fly balls were called the high-pressure governor by the engineers.

I wanted to know if, in case of overspeed, the engineer or oiler could operate the valve. I was told yes, and that the automatic stop could be used instead of the throttle at shutting-down time.

I suggested that we try it at the first chance, and we did. Did the automatic safety-device work? Sure it worked, but not until we had spent several days scraping off burned oil and repacking.

G. D. Crain says in the same issue that "owners of larger plants have their boilers insured, which results in high-grade inspection." I beg to differ. I know of boilers that were insured and regularly inspected and just as regularly reported in good condition; I also know that many of the tubes were packed solid with scale.

The editorial already mentioned says, "He is a wise engineer who will use those safety devices which lend themselves to hand as well as automatic control in handling the machinery which they should safeguard."

He also is a wise engineer who will inspect his own boilers, no matter how many inspectors may do it besides.

Chicago, Ill. W. H. Macking.

Inquiries of General Interest

Grate Area with Mechanical Stokers—For a given boiler capacity, why is less grate area required for mechanical stokers than for hand-fired grates?

H. C.

The motion of the mechanical stoker maintains a cleaner fire by continuously disturbing the film of ash which is formed on the burning coal, so that the rate of combustion is greater per square foot of grate.

Causes of Pump Running Lame—What causes a duplex pump to run lame—i.e., the strokes on one side are shorter in length of time than those of the other?

B. L.

The slower-running side may have a leaky steam piston or the resistances of the water cylinders may be different, as from a worn pump valve, a leaky water piston or leaky valves on the quicker-running side or tighter water piston or stuffing-boxes on the slower-running side. When such faults have been corrected the lost motion of the steam valves should be adjusted to obtain the desired equality of strokes.

Using Soda Ash for Removal of Scale—Will a boiler be injured by using soda ash in the following manner? Before washing the boiler put in about 40 lb. of the soda ash, again close the boiler, and with the water a little above working level, boil slowly for a few hours, and after allowing the boiler to stand for three or four days, wash thoroughly.

R. G. T.

There should be no injurious results if care is taken to thoroughly wash out all traces of sludge from the try-cock and water-column connection and the soda solution is not permitted to enter the safety valve, for upon drying it would be likely to cement the valve to its seat.

Pipe Sizes for Hot-Water Heating—What is the rule for determining the sizes of supply and return pipes for a gravity hot-water heating apparatus with direct radiation?

J. M.

As there is no sensible change in bulk between the supply and the return water, the supply and return pipes should be of the same diameter. With ordinary conditions, and where the supply or return pipe is less than 200 ft. in length, it is a good practical rule to allow one pipe size greater than the square root of the number of square feet of radiating surface, divided by 9 for the first story, by 10 for the second story, and by 11 for the third story of a building.

Relative Volumes of Water and Steam—What are the relative volumes of a pound of water at 212 deg. F. and a pound of steam at atmospheric pressure?

W. R.

A cubic foot of water at 212 deg. F. weighs 59.833 lb., and one pound occupies a space of

$$\frac{1}{59.833} = 0.0167 \text{ cu.ft.}$$

while one pound of dry saturated steam at atmospheric pressure (14.7 lb. per sq.in.) occupies a space of 26.79 cu.ft., which is

$$\frac{26.79}{0.0167} = 1604.19 \text{ times the space occupied by the water.}$$

Indicated, Brake and Friction Horsepower—What is meant by indicated, brake and friction horsepower of a reciprocating engine?

S. C. M.

Indicated horsepower (abbreviated i.hp.) is the power delivered to the piston by the steam or other working fluid which is employed for moving the piston, and is so called because the effective pressure is usually determined by use of a steam-engine indicator. Brake horsepower (abbreviated b.hp.) is the power delivered by the engine exclusive of the power wasted in overcoming the friction of its moving parts. The brake horsepower is therefore always less than the indicated horsepower, the difference being the power required to overcome the friction of the engine. This difference is sometimes called the friction horsepower.

Capacity of Triplex Plunger Pump—What is the capacity in gallons pumped per minute of a single-acting triplex plunger pump having plungers 5½ in. diameter by 8 in. stroke and making 28 r.p.m.?

M. M. G.

The cross-sectional area of each plunger would be 5½ × 5½ × 0.7854 = 23.758 sq.in., and having three plungers, each with 8-in. stroke, the total displacement would be:

$$23.758 \times 8 \times 3 = 570.192 \text{ cu.in. or}$$

$$570.192 \div 231 = 2.468 \text{ gal. per rev. or}$$

$$2.468 \times 28 = 69.1 \text{ gal. per min.}$$

The actual amount of water pumped would be less according to the "slippage," which would depend on the temperature of the water, height of suction lift, size, design and arrangement of suction and discharge valves and piping, and adjustment of the plunger packing. Under ordinary conditions the slippage would amount to about 5 per cent. and the net pumpage would be 95 per cent. of 69.1 or about 65½ gal. per min.

Racing of Electric Elevator—What would cause an electric elevator to race occasionally on its upward trip?

J. B.

If the electrical connections are not making good contact and the shunt field circuit is open, the motor will race, if compound-wound, or if shunt-wound the fuse will blow. A ground in the field coils, combined with a ground on some other part of the system, will also cause high speed. If the car is over counterweighted it may race on the up travel if the series field winding is not cut out of circuit. This is accomplished when the starting resistance is cut out, consequently it may be that the rheostat arm has stuck. Again, the operating mechanism may have drifted back to where it cut off the current from the motor, but not far enough to apply the brake. This would cause the car to race on the up travel without load if the counterweights are heavier than the car. Also, if the controller has been recently overhauled, some of the connections may have been misplaced.

Copper-Ball Pyrometer—How is the temperature of gases escaping from a boiler determined by means of a copper-ball pyrometer?

R. G.

A copper ball of known weight is suspended in the uptake at a point where the temperature is to be taken and after the ball has attained the same temperature as the surrounding gases it is quickly dropped into a vessel containing a known weight and temperature of water. The water is rapidly stirred and its maximum temperature is taken, i.e., when the ball and water have attained the same temperature. The temperature of the copper ball before being cooled would then be given by the formula:

$$x = \frac{W\,(T - t)}{w\,S} + T$$

in which

x = Temperature sought;
W = Weight of the water, in pounds;
w = Weight of the copper ball, in pounds;
t = Initial temperature of the water;
T = Maximum temperature of the water and final temperature of the copper;
S = Specific heat of copper, which may be taken as 0.095.

For example, if the weight of water is 20 lb., the weight of copper ball 12 lb., the initial temperature of the water 52 deg. F. and the maximum temperature of the water and final temperature of the copper is 85 deg. F., then the temperature of the copper ball before being cooled would be

$$\frac{20 \times (85 - 52)}{12 \times 0.095} + 85 = 663.9 \text{ deg. F.}$$

The actual temperature of the waste gases will be somewhat more than the result obtained by use of the formula, on account of corrections for variations in the specific heat of the water and metal for different temperatures, losses of heat by radiation of the metal during transfer from the uptake to the water, and heat lost during the heating of the water and absorbed by the vessel containing it.

[Correspondents sending us inquiries should sign their communications with full names and post office addresses. This is necessary to guarantee the good faith of the communications and for the inquiries to receive attention.—EDITOR]

Engineers' Study Course

The Efficiency of Heat Engines

A heat engine is an apparatus for converting heat energy into mechanical work. In engines of the internal-combustion class this whole process is carried out within the self-contained machine to which the name "engine" is commonly given. In a steam plant functioning begins at the heating surface of the boiler (where heat is received) and extends to the condenser (where heat is given up), so that here the term "heat engine" covers a good deal more than the steam engine or turbine alone.

The efficiency of any energy converter is the ratio of useful output to total input. There is a class of transformations—typically, those of the electric generator and motor and of machinery transmitting power—in which output is, or may be, nearly equal to input. The difference (decrease) is due to secondary losses, which may be controlled and diminished, but never wholly eliminated. However, it is easy to imagine a perfect electro-dynamo, or a frictionless machine, which shall have no losses and, therefore, unit efficiency. This ideal action, with the apparatus delivering in useful form all the energy that it receives, stands as a limit of performance to which the actual machine approaches more or less closely.

But the heat engine does not tend to approach unit efficiency or complete conversion of heat as its action is improved. Instead, there is a limiting efficiency of lower value, which depends upon the general conditions of operation and is expressed by a fraction ranging in various cases from 0.15 to perhaps 0.65. The typical range for good steam plants, running condensing, is 0.30 to 0.35—this being in terms of heat received by the steam from the fire and, therefore, not including boiler efficiency. In other words, an ideally perfect steam-engine plant, with losses by reason of radiation, cylinder-wall action, or resistance to flow of steam, could convert into work 30 to 35 per cent. of the heat received, and would unavoidably reject in its exhaust the remaining 70 to 65 per cent. The actual plant will do about two-thirds as well as the ideal, converting 20 to 24 per cent.

Having given nothing more than the general law or principle that energy is convertible from one kind to another, one might think that the conversion, for instance, of heat into work could be effected in a number of ways, of which the best or most convenient would be chosen for practical application. Really, however, no such freedom of choice exists with this particular conversion; rather, there is but one general method available, that which depends upon the use of an expansive medium, a vapor or gas. Of course, attempts have been made to find or invent some other way, but these have been altogether fruitless, and the heat engine of common type is the only known device for getting work from heat.

The question now to be considered is, "How is it that the heat engine, even under ideal conditions, cannot have unit efficiency?" The word "how" is used instead of "why," of intention. The latter might imply that the question was one to be solved by some train of abstract reasoning based upon an initial concept of the general nature of energy. On the contrary, there must be study and analysis of the physical processes involved, bringing them to expression in simplest terms and leading to a simple final statement of physical impossibility.

In the manner just indicated let us now consider two typical heat engines, taking the steam plant with piston engine and the gas engine with explosive or Otto cycle as representative examples. Their cycles of operation, briefly described in parallel schedules, are as follows:

1. Taking the charge and bringing it to the state where it begins to receive heat.

Steam plant: For each working cycle of the engine, a certain amount of water under atmospheric pressure and at a temperature not higher than the atmospheric boiling point is put into the boiler by the feed pump, which performs the work necessary to that end.

Gas engine: A cylinderful of mixed air and gas is drawn in and then compressed. In the ideal case the charge would neither receive nor yield up any heat during this operation, so that the compression would begin at atmospheric temperature and would be truly adiabatic.*

2. Imparting heat to the charge.

Steam plant: The water is first raised to the temperature of steam formation and then vaporized, the latter effect requiring by far the larger part of the heat supplied. Room for the very great increase of volume from liquid to vapor is made by the advance of the engine piston, out to cutoff. The work of this expansion is, of course, performed usefully upon the piston.

Gas engine: The charge is ignited and its heat of combustion causes a great rise of temperature and pressure, the volume remaining nearly constant while the piston is moving slowly near dead-center.

3. Expansion without heat supply.

Steam plant: After cutoff, the steam expands behind the advancing piston.

Gas engine: This expansion constitutes the whole of the working stroke.

In both cases the ideal condition for best effect would be to have this expansion take place in a cylinder that was thermally neutral, or that would neither conduct, absorb, nor give up heat. The unavoidable departure from adiabatic action, with real metal cylinders, is one of the chief causes of the gap between ideal and actual performance.

4. Exhaust and removal of unused heat.

Steam plant: Except for a small amount of radiation, the heat not converted goes out in the exhaust steam. Whether taken into the atmosphere directly or into a current of condensing water, it ultimately settles down to the general level of outside temperature.

Gas engine: Heat is taken from the cylinder continually by the jacket water, and the rest of the unconverted heat goes out into the atmosphere in the exhaust gases.

The purpose of the preceding brief review of well

*Adiabatic is a short equivalent for the phrase, "without giving or receiving heat."

known facts is to furnish a reason for, and a practical illustration of, the following general statements:

To get work from heat it must be available at some high temperature, well above that prevailing in the atmosphere and other surrounding bodies.

This heat is applied to some liquid or gaseous medium, causing it to expand and do work.

When at the end of expansion, which is limited by its falling to a pressure below which no useful effect can be got, the medium still contains a large part—commonly much the larger part—of the heat which it has received. Nothing can be done with this heat but to reject it to the atmosphere or surrounding bodies.

All of the work done by the steam or gas in its expansion is not useful output, because the piston must give back some work in expelling the exhaust or compressing the new charge.

These statements are more nearly related to the actual practical engine than to the abstract and, in several respects, imaginary apparatus which is assumed in thermodynamic discussions. They leave some loose ends, but cover the ground well enough. To sum up, the reason why the heat engine cannot attain unit efficiency is that at the end of expansion, when the working medium has performed all the work of which it is capable, it still contains a large portion of the original supply of heat, which has been reduced in temperature and can only be given up or thrown away at this low temperature.

In connection with the performance of any heat engine there are three efficiencies to be considered, which can most clearly be defined in relation to an example. The best test of the Brown-Boveri-Parsons turbine at Newcastle-on-Tyne, which was reported in POWER for Apr. 18, 1911, gave the following data and results:

Load, 6257 kw., on a rating of 6000 kw.; steam pressure, 204 lb. abs.; steam temperature, 560 deg. F., or 176 deg. of superheat; exhaust (condenser) pressure, 0.44 lb. abs., corresponding with a steam temperature of 76 deg.; steam consumption, 11.95 lb. per kw.-hr.

Since one kilowatt-hour is equivalent to 3412 B.t.u., the heat energy converted and delivered at the busbars for each pound of steam is

$$3412 \div 11.95 = 286 \ B.t.u.$$

The heat of formation of one pound of steam, assuming feed water at the temperature of the exhaust steam, is

$$1299 - 44 = 1255 \ B.t.u.$$

With ideal Rankine-cycle performance, the output per pound of steam would be 416 B.t.u. converted into work. Then the ideal efficiency is

$$\frac{Ideal\ output}{Heat\ input} = \frac{416}{1255} = 0.332$$

The actual, absolute efficiency is

$$\frac{Actual\ output}{Heat\ input} = \frac{286}{1255} = 0.228$$

And the relative efficiency, the ratio of actual to ideal performance, is either

$$\frac{Actual\ efficiency}{Ideal\ efficiency} = \frac{0.228}{0.332} = 0.687, \text{ or}$$

$$\frac{Actual\ output}{Ideal\ output} = \frac{286}{416} = 0.687$$

This ratio is the real criterion of effectiveness, since it shows how well those losses which are more or less subject to control are kept down. Of course, the 0.228 and 0.687 include the combined mechanical and electrical efficiency of the turbo-generator, which is probably about 0.96. Therefore, the efficiencies in terms of power developed by the steam on the turbine rotor are probably about 0.24 and 0.72.

As Dangerous as Dynamite

A steam boiler with an inoperative safety valve is about as dangerous as a supply of dynamite. Wagons loaded with high explosives may be seen any day passing through the streets of crowded New York. Attention, however, is drawn to the danger by means of large red letters and and a red flag hung conspicuously from the rear of the vehicle. But boilers carrying high steam pressure and having plugged safety valves exist under the sidewalks of this great city without any warning visible to the unsuspecting pedestrian.

This statement may seem sensational, but it is based upon fact.

Just recently an insurance company was asked to insure a boiler carrying a pressure of 30 lb. After examining the boiler the inspector ordered the pressure reduced to 15 lb., but the owner wanted to carry 30 lb., and upon his earnest request the insurance company consented to make a re-examination. When the inspector arrived he found the safety valve blocked, a stick of wood having been wedged between the top of the valve lever and the bottom of the ceiling joist in the building. Needless to say, the insurance was immediately suspended. Insurance on this boiler would not now be accepted even after the pressure had been reduced and the safety valve unlocked.

Fortunately, it is a rare occurrence to find a safety valve that has been purposely blocked to permit the carrying of higher steam pressure. In fact, the writer knows of but one other case, and that was done by a negro fireman who stated that he had to do it to prevent the safety valve from discharging steam.

Safety valves, however, are often blocked unintentionally or by accident. Not long ago a boiler located under a Broadway sidewalk at Forty-first St., after undergoing repairs, was tested by hydrostatic pressure. To apply this pressure it was necessary to block the safety valve. The boiler attendant, unaware that the boiler tester had not removed the block, raised steam on the boiler. The boiler exploded, badly damaged the surrounding property and fatally injured the son of the owner.

In another case the corrugated ceiling over two boilers sagged to such an extent as to come into direct contact with the top of the safety valves. The boilers were located under the sidewalk of a crowded thoroughfare. Fortunately, the danger was discovered by an inspector before explosion occurred.

All but one of the instances here cited occurred in New York City within the last two or three months. Considering the vast number of boilers throughout the United States, it is reasonable to suppose that a very considerable number are operated under dangerous conditions.

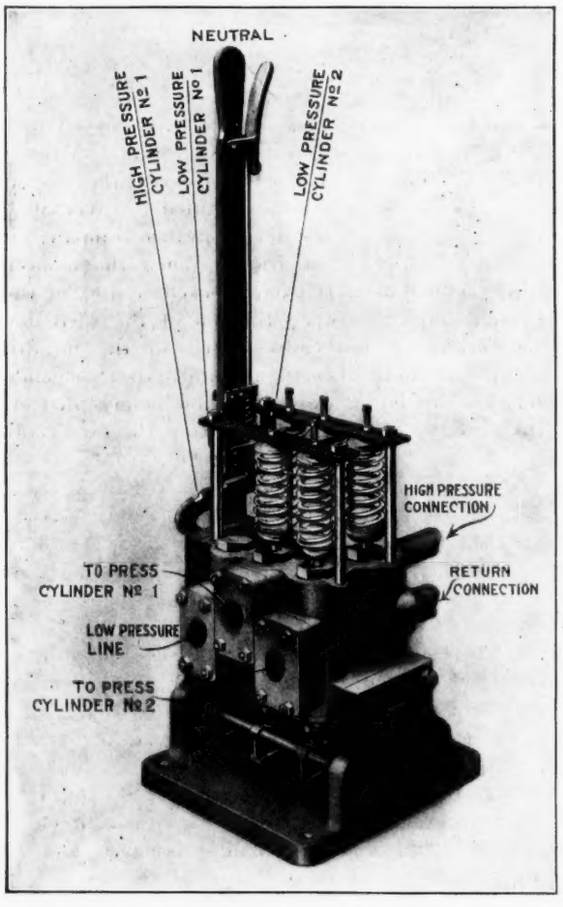
It is obvious, therefore, that frequent inspections by trained experts are absolutely necessary. There is reason for the movement recently inaugurated in so many states for the adoption of a uniform compulsory boiler-

inspection law—a law which will provide for regular inspections, either by state inspectors or by the inspectors of boiler-insuring companies duly authorized to insure boilers in the given state.—*Monthly Bulletin of the Fidelity and Casualty Co., Jan., 1915.*

New Hydraulic Valve

Operators of hydraulic equipment of the double-acting character have experienced difficulty in obtaining proper pressure control with their valve equipment when the ram is forced in both directions by hydraulic pressure.

To meet this demand the Hydraulic Press Manufacturing Co., Mount Gilead, Ohio, has designed the five-way high- and low-pressure double-acting balanced poppet operating valve illustrated herewith.



Five-Way High- and Low-Pressure Double-Acting Balanced Poppet Operating Valve

The low pressure is admitted to the first cylinder, leaving the second cylinder open to the return line. When the low pressure has done its work in the first cylinder, the high pressure is turned on. A check prevents the liquid from the high-pressure line from flowing into the low-pressure line. The valve can then be shifted to the position which applies low pressure to cylinder No. 2 and releases cylinder No. 1. A similar valve is made with another position, which applies high pressure to cylinder No. 2 with No. 1 still open. In most cases the latter position is not necessary, as the work of cylinder No. 2 is done at low pressure only, as in the case of auxiliary return cylinders. On account of the length of the operating lever it is necessary for the operator to stand above the level upon which the valve rests.

The valve has five stems and checks and is suitable for use with pressures up to 5000 lb.

Appearance as an Element in Power-Plant Value

By Edwin D. Dreyfus

Doubtless the mentioning of the fact that the appearance of and the general care accorded a power plant or other operating system possess concrete value will at first seem commonplace. Everyone will contend that he realizes the importance of good order in any working institution. But do the majority of us take sufficiently seriously the slogan of "watchful care and attention" and make the necessary effort to keep our house in the very best order both from an interior and an exterior standpoint? We may neglect the equipment entrusted to our care so long as no accounting is required, but let the occasion arise and we will quickly find means to do vastly more than we had previously attempted. As it is inherent in us to accomplish greater things, why do we not assume the initiative rather than have the doing of these things urged upon us?

A new era has dawned wherein we find ourselves in a condition of strict regulation, either by keen competition or else through municipal, state or federal supervision. The "survival of the fittest" is going to be more pronouncedly the byword in the future, and it is with this in view that the writer attempts to point to instances wherein the orderliness of the plant may represent an indirect monetary value of material consequence.

Regulation is the order of the day—mainly that of public-service corporations, but to an increasing extent of industrial companies doing an interstate business as well—and though there may be other angles by which this is approached, price or rate control is the one of immediate interest. To determine the proper prices for the output of any plant, we must fix upon a reasonable return upon the investment in or the value of the plant. We need not concern ourselves in this article with what this should be as it is governed by the financial risk. However, the fair value is another matter, and should not be a variable quantity, as it is dependent upon the legitimate investment. The book records of the operating company should show this, but it is only recently that records have been kept that exhibit the construction cost separate from other charges.

Consequently, to derive the value as of a certain date, an appraisal must be made in the majority of cases. Herein lies the tangible value of the general appearance of the plant, because some courts and commissions hold that the present value shall be determined by the "reproduction cost new less depreciation" method. Evidently, if the plant gives an impression of neglect, the deterioration and consequent depreciation may appear more exaggerated than they actually are. Whatever diminution in value may result from the inspection (presuming the regulatory body to be guided strictly by the investigator's report) will in all probability represent a permanent loss unless an appeal is made by the company affected.

It is hardly to be disputed that the company's and the employees' interests are interwoven. If the company possesses an efficient management the welfare of the

employee is enhanced, and if the employees are individually efficient the company is insured of greater prosperity.

Let us attempt to illustrate the relationship numerically, taking the power plant as an example. Suppose it is to be appraised for price or rate regulation. Assume that the normal rated capacity is 1000 kw. and the cost to install was $100 per kilowatt, or a total of $100,000. Say where the plant was well maintained the condition was found to be 85 per cent. and where indifferent attention obtained the property was considered to be worth 75 per cent. of its original value of $100,000. Accordingly, the less industrious engineer would have lost his employer 10 per cent., or $10,000. Now if the company had been allowed to earn 8 per cent. upon the present value of its property devoted to public interest and to set aside 5 per cent. to cover necessary renewals and replacements, together with 1 per cent. for insurance and taxes, it should receive 14 per cent. after operating expenses have been met. Through the writing down of the 10 per cent. greater depreciation in one case than in the other the company's earnings may have been reduced 14 per cent. of $10,000, or $1400 per year. And this in a larger degree represents the value of the ambitious engineer-in-charge over the one less energetic.

Volumes have been written upon the different methods of arriving theoretically at depreciation. During the early stages of regulation the "life" method principally was in vogue. Now the theoretical method has substantially given way to determining the amount by actual inspection. Upon this development rests the fact that the plant appearance, irrespective of the running condition, becomes of intrinsic value.

By the life, or theoretical, method the depreciation was arrived at by taking the average length of service obtained from machinery and property as was shown by a wide range of experience. Such results had to be predicated upon the performance of equipment both carefully and carelessly maintained, and consequently this method worked an injustice in some cases and disproportionately favored others. Hence, we have come to employ the actual inspection by a competent person in virtually all recent work.

Inspection to determine the reduction in value, like a great many other things in our daily life, is not an exact science, and in this field particularly the psychological influence and mental impression produced may play a considerable part in the results.

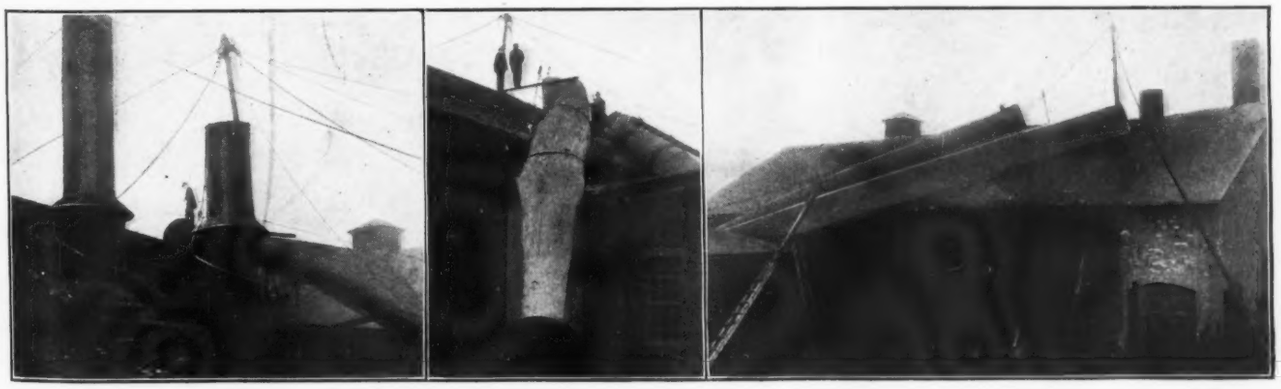
To a great many this discussion may seem mere platitude, but the writer has been in many plants where there was apparently no appreciation of the value of appearances. It is not uncommon to find engineers studying indicator cards, analyzing flue gases and carefully observing the general load and operating conditions, but otherwise unmindful of orderliness. The man who not only secures the last fraction of economy and high efficiency but maintains orderly arrangement is going to be the most highly valued. I have known of two engines which were both running satisfactorily, but the older one was the more carefully groomed and was accordingly marked down for far less depreciation.

In enumerating what could be done in a power plant to maintain a good appearance it might be mentioned that in boiler settings all cracks should be calked and the brickwork pointed up; in rotating machinery all side play and any wabbling due to sprung shafts should be eliminated, although having no bad effect upon the operation of the machinery; pumps in the boiler room subject to accumulation of ash and coal dust caked with grease should be kept rubbed down; boiler fronts kept in brightened condition; water- and steam-pipe leaks stopped even where they are of trifling consequence from an economical standpoint; and the tools and all material and supplies arranged in order.

The old adage that "Cleanliness is next to godliness" is thereby a virtue growing in significance. It is of course understood that "surface" conditions alone will not be sufficient, but the internal state of repairs is sometimes more important. However, it is quite clear that outside appearances virtually have a tangible value.

Dead-Man Broke

On Monday, Jan. 4, at about 3 p.m., a peculiar accident occurred at No. 2 pumping plant of the Jamaica Water Supply Co., Jamaica, Long Island, N. Y. The erection of two new smoke-stacks had just been completed, and the riggers were preparing to leave the premises when the guy stub, or dead-man, broke off near the ground, allowing the stacks to fall over. The dead-man was a short length of railroad rail, probably steel, set into the ground. At the point of failure it showed no bend whatever, but simply a short break.



CONDITION OF STACKS WHICH FELL WHEN GUY STUB OR DEAD-MAN BROKE

By prompt and wise action on the part of the chief engineer and the riggers, the service was not interrupted and the output was but slightly reduced for a short time only. The rivets were cut at a joint five or six courses from the bottom, and these short stacks set up in their original places, as shown in the illustration. By this means the boilers were operated by forced-draft blowers while the other sections of the stacks were repaired and made ready to replace.

Although there were several men near-by, no one was hurt.

Boiler Explosions During First Half of 1914

Date	Place	Plant	Time of Day	Kind of Boiler	Pressure Carried, Lb.	Size	Killed	Injured	Property Loss	Probable Cause
Jan. 1	St. Louis, Mo.	Martin-Haloran-Klaus Laundry Co.	12 M.	Water-tube	125	75 hp.	1	2	$100	Tube ruptured
Jan. 1	Toledo, Ohio	St. Anthony Church and School		Cast iron			0	0		Low water
Jan. 2	Buffalo, N. Y.	Erie R.R. Loco.		Locomotive			2	0		
Jan. 2	Indianapolis, Ind.	Arcadia Dancing Academy					0	0	$500	
Jan. 2	Lower Glenwood Landing, Penn.	Pittsburgh Coal Co.			115		0	0	$1000	
Jan. 2	Terre Haute, Ind.	T.-H., I. & Eastern Traction Co.		Water-tube	160		0	0		Cast-iron header ruptured
Jan. 3	Muskogee, Okla.	C. W. Raymond		Cast-iron sectional					$75	
Jan. 4	Norwood, N. Y.	Norwood Elec. Light Plant		Return-tubular	125	150 hp.	0	0	$1000	Unknown
Jan. 4	Lima, Ohio	Ohio Elect. Light Co.		Water-tube						Cast-iron header ruptured
Jan. 4	Dawson Springs, Ky.	New Century Hotel		Cast-iron heating			0	0		Section cracked
Jan. 5	Denton, Tex.	State Normal School		Cast-iron heating						
Jan. 5	Countiss, Ark.	Hammonds Lumber Co.					1	5		
Jan. 5	Switchback, W. Va.	Appalachian Paper Co.		Water-tube			0	1		Tube rupture
Jan. 7	Pelham, Ga.	Pelham Oil and Fertilizer Co.					0	0		
Jan. 7	Independence, Kan.	Carl Leon Hotel		Cast-iron heating	10	Mod. 40-11	0	0		Five sections cracked
Jan. 7	East Liverpool, Ohio	Tri-State Railway & Elec. Co.	8:00 p.m.	Water-tube			0	0		(Two tube failures)
Jan. 7	North Cambridge, Mass.	Willowcraft Shop		Cast-iron heating	15		0	0	Rep. by mak'r	Defect in casting
Jan. 8	Fisher, Ill.	Earl Campbells Farm	8:00 a.m.	Locomotive type	140	15 hp.	0	2	$200	Heads of staybolts corroded off
Jan. 8	Bridgeport, Ill.	Ohio Oil Co.	8:00 a.m.				3	4	$2500	
Jan. 9	Lowell, Mass.	J. N. Adam's Dept. Store					0	0		Blow-off failed
Jan. 9	Lowell, Mass.	Prescott Mills					0	0		
Jan. 10	Shaverstown, N. Y.	Bartlett Bros. Sawmill	9:30 a.m.	Cornish	100	25 hp.	2	0	$300	Unknown
Jan. 10	New York City (Astoria)	Malchay & Gibbons, Contractors	12 M.	Upright-tubular			0	16		
Jan. 10	Terre Haute, Ind.	T.-H., I. & Eastern Traction Co.		Water-tube type	160		0	0		Tube rupture
Jan. 11	Cadillac, Mich.	Commer-Diggins Mill					0	2		Low water
Jan. 12	East Lebanon, Penn.	Leb. Val. Iron & Stl. Co.					0	5		
Jan. 12	Kansas City, Mo.	Moore Hotel		Cast-iron heating			0	0		
Jan. 12	Waukegan, Ill.	American Steel Wire Co.		Water-tube			0	0		Cast-iron headers ruptured
Jan. 12	Bridgeport, Ill.	Standard Oil Co. Oil Lease	9:00 a.m.	Locomotive type	120	25 hp.	3	4	$500	Low water, starting injector
Jan. 12	Albany, Ga.	Drake & Thompson Sawmill	7:00 a.m.	Return-tubular	100	100 hp.	1	2	$3000	Valve closed between boiler and safety valve
Jan. 12	Nowata, Okla.	C. Newbert's Well-Driller	8:00 p.m.	Locomotive type	120	abt. 30 hp.	0	2	Boiler only	Low water
Jan. 12	Columbia, S. C.	G. A. Guignard's Brickyard		Locomotive			0	1		Crown sheet failed
Jan. 13	Rochester, N. Y.	34 Concord St.					0	0		
Jan. 13	New York City	Ansonia Apartment Hotel	1:00 a.m.				0	1		Valve burst
Jan. 13	Flat Rock, Ill.	Ohio Oil Co.	11:00 a.m.				0	1		
Jan. 13	Shirley, Ark.	Smile Lumber Mills	3:00 p.m.				2	0		Cold water admitted to empty boiler
Jan. 13	Braddock, Penn.	American Steel & Wire Co.		Water-tube			0	0		Cast-iron mud drum failed
Jan. 13	Roxbury, Mass.	Apt. No. 6 Humboldt Ave.		Cast-iron heating	15	9 sections	0	0	Repl'ing sec.	Section cracked
Jan. 13	Lima, Ohio	Ohio Elect. Railway Co.		Water-tube			0	0		Four cast-iron headers failed
Jan. 13	Lawrence Co., Ill.	George Gray Farm	11:00 p.m.		100	25 hp.	0	1		Low water
Jan. 14	Portchester, N. Y.	119 Horton Ave.					0	0		
Jan. 14	Philadelphia, Penn.	J. Block					0	3		
Jan. 14	Red Bank, N. J.	Santangelo's Bottling Wks.					0	0		
Jan. 14	Jersey City, N. J.	619 Summit Ave.					0	0		
Jan. 14	Jersey City, N. J.	27 Palisade Ave.					0	0		
Jan. 14	Philadelphia, Penn.	F. M. Worstell					0	0		
Jan. 14	Philadelphia, Penn.	29 Noble Street					0	1		
Jan. 14	Wellsburg, W. Va.	Gas Plant					0	Several		
Jan. 14	Coshocton, Ohio	Coshocton Light & Heat Co.	10:00 a.m.	Water-tube	140	150 hp.	0	1	Tube	Tube rupture
Jan. 14	North Adams, Mass.	J. Alpert		Water front of stove			2	0		Frozen pipe connection
Jan. 14	Jersey City, N. J.	Gotthelf's Store		Kitchen boiler			0	3	$1000	Service pipe frozen
Jan. 14	Huntington, Penn.	Fred Etnier's Home		Cast-iron heating			0	0	$3500	
Jan. 14	Plainfield, N. J.	A. Hirsch's Home		Cast-iron heating			0	0		
Jan. 15	Butte, Mont.	Pittsmont Mine					0	0		
Jan. 15	Boston, Mass.	Apt. House 98 to 102 Gainsboro St.		Cast-iron heating			0	0		
Jan. 15	Holyoke, Mass.	Garage, E. H. Friedrich					0	1		
Jan. 15	Beaverton, Ala.	Jim Black's Sawmill	8:00 a.m.	Return-tubular	100	20 hp.	3	0	$1000	Low water and water admitted
Jan. 16	Long Island City, N. Y.	J. F. Blanchard		Return-tubular	125		0	0	$100	
Jan. 16	Littleton, N. H.	Saranac Glove Co.		Return-tubular	70	100 hp.	0	0		Crack in sheet. No explosion
Jan. 17	Federalsburgh, Del.	Long Bro's & Co. Store	3:00 p.m.	Cast-iron heating	5	40 hp.	0	0		Low water
Jan. 17	Point Pleasant, W. Va.	Grist Mill	2:00 p.m.	Locomotive type	85	60 hp.	0	2	$460	Pressure gage out of order
Jan. 17	Lancaster, Penn.	Lefevre's Greenhouse					0	0		
Jan. 18	Galena, Mo.	Campbell Bros. Mine	12:00 p.m.				0	0		
Jan. 18	Mt. Pleasant, Ia.	State Hospital for Insane		Water-tube			0	0		Tube ruptured

BOILER EXPLOSIONS DURING FIRST HALF OF 1914—Continued

BOILER EXPLOSIONS DURING FIRST HALF OF 1914—Continued

Date	Place	Plant	Time of Day	Kind of Boiler	Pressure Carried, Lb.	Size	Killed	Injured	Property Loss	Probable Cause
Jan. 19	Haverhill, Mass.	W. & V. O. Kimball Co.					0	0	Repairs	Leak only, no explosion
Jan. 20	Lynn, Mass.	J. B. Blood & Co. (Grocery House)		Cast-iron sectional			0	0	Repl'ing sec.	Seven sections ruptured. Low water
Jan. 20	Ft. Wayne, Ind.	Ft. Wayne Bldrs. Supply Co.			90		0	0	$25.00	
Jan. 21	Feehansville, Ill.	St. Mary's School for Boys					0	0		
Jan. 21	Warren, Ill.	High School	9:15 a.m.	Cast-iron heating	3	S. 36-8	0	0	$100	Four sections cracked
Jan. 21	Ottawa, Ont., Can.	Horwick Hall	8:05 a.m.	Return-tubular	20-40	60 in. x 14 ft.	3	6	$100,000	Safety valve inoperative
Jan. 22	Des Moines, Ia.	B. F. Kauffman & Office Bldg.					0	0		
Jan. 22	Shelby, Ala.	Shelby Iron Co.					2	1	$2000	
Jan. 22	El Paso, Tex.	El Paso & Southwestern Railway					1	3		
Jan. 22	Elyria, Ohio	Clev., Southwestern & Col. Trac. Co.		Water-tube	150	425 hp.	0	1	Tube	Tube pulled out
Jan. 23	Kewjawhatchee, Ala.	Shelby Iron Wks.					1	0		
Jan. 23	Washington, D. C.	The Chapin-Sacks Mfg. Co.	11:40 p.m.	Water-tube	140	250 hp.	0	1	$300	Tube ruptured. Defective tube
Jan. 26	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	150 hp.	0	0		Cast-iron header failed
Jan. 26	Gloversville, N. Y.	104 Water St.					0	0		
Jan. 26	Kansas City, Mo.	Charlotte Apartments		Hot water			0	0	$150	Defective connections
Jan. 27	Pittsburgh, Penn.	National Tube Co.		Water-tube			0	4		Tube ruptured
Jan. 28	Omaha, Neb.	Y. M. C. A.	11:00 a.m.	Return-tubular	100	125 hp.	0	0	$100	Mud and scale
Jan. 30	St. Augustine, Fla.	Sarasota Ice & Elect. Co.					0	0		
Jan. 31	Lovell, Wyo.	Byron Oil Well		Portable	100		0	0	$1000	Crown sheet failure. Stay bolts all stripped
Jan. 31	Chicago, Ill	Union Special Machine Co.					1	0		Flues failed
Feb. 2	Miami, Fla.	Johnson & Co. (Dredgeboat)					0	0		Crown sheet failed
Feb. 2	Bakersfield, Calif.	Southern Pacific Locomotive	3:20 p.m.	Locomotive (Mallet)			2	0		Unknown
Feb. 3	Lowell, Mass.	Owl Amusement Co.'s Theater		Cast-iron sectional			0	0		Two sections ruptured
Feb. 3	Utica, N. Y.	Hotel Utica Garage	6:00 a.m.	Blow-off Tank			1	1		Outlet closed
Feb. 4	Shelby, Ala.	Iron Works Pumping Station					1	1		
Feb. 5	Urban, Ky.	Sawmill	3:00 p.m.	Locomotive type	110	45 hp.	6	4	$1000	Low water, cold water put in
Feb. 5	Howenstine (near) Canton, Ohio	Traction Engine			150		1	3		Overpressure
Feb. 6	Waukegan, Ill.	Am. Steel & Wire Co.		Water-tube			0	0		Tube ruptured
Feb. 6	Hiawatha, Kan.	Brier Shadel Mercantile Co.		Cast-iron sectional	30' head		0	0	$50	Scale, hot water used from system
Feb. 6	Indianapolis, Ind.	Indianapolis Light & Heat Co.		Water-tube	200	612 hp.	0	0		Water hammer, strained rivets in head
Feb. 7	St. Louis, Mo.	Wabash Locomotive	4:00 p.m.				0	3	$750 to bldgs	Crown sheet failed
Feb. 8	Canal Dover, Ohio	Reeves Mfg. Co. Tin Mill		Water-tube			0	0		Tube ruptured
Feb. 9	St. Louis, Mo.	Union E. L. & P. Co.		Water-tube	175		0	0	$140	
Feb. 9	Ft. Wayne, Ind.	Tompkinson Plumbing Shop					0	1		
Feb. 9	Coronado, Calif.	Coronado Beach Co.		Water-tube			0	2		Tube ruptured
Feb. 11	Dubuque, Ia.	Wm. Lawther Co.'s Plant	2:30 a.m.	Return-tubular	70	52"x16'	0	0	$100	Blow-off failure
Feb. 11	Bakersfield, Calif.	Bankline Oil Co. (Pumping Plant)					0	0		
Feb. 12	McComb, Ohio	Smith Block					0	1		
Feb. 12	Asbury Park, N. J.	North Asbury Fire Co. Station		Heating			0	0		
Feb. 12	Arlington, Mass.	Arlington Five Cent Savings Bank		Cast-iron sectional	8		0	0	Repl'c'g sect.	Low water. Section cracked
Feb. 12	Wellsley, Mass.	Dana Hall School		Cast-iron heating	7	12 sections	0	0	$500	One section and one manifold failed
Feb. 12	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	150 hp.	0	0		Cast-iron header ruptured
Feb. 12	Cleveland, O.	J. M. Gasser & Co. (Greenhouses)		Fire-tube	60	75 hp.	0	0		Blister in firebox caused by scale. No explosion
Feb. 12	North Bergen, N. J.	No. 5 School Building					0	3		
Feb. 12	Ravana, N. Y.	Grammar School	10:00 a.m.	Cast-iron heating		3500' cap	0	0		Old boiler worn out
Feb. 12	Burlington, N. J.	Street Car Heater					0	4		
Feb. 16	Cedar Rapids, Ia.	J. A. Moeller's Creamery					0	1		
Feb. 16	Cincinnati, Ohio	Cincinnati Coffin Co.					0	1		Blow-off failed
Feb. 16	Chicago, Ill.	P. & S. R.R. Demonstration Car					0	1		
Feb. 16	Detroit, Mich.	Fredk. Sterns & Co.					0	1		Blow-off failed
Feb. 16	Knob Noster, Mo.	Acme Milling Co.	6:45 a.m.	Horizontal-tubular	85	16x54	0	0	$1000	Low water
Feb. 16	Terre Haute, Ind.	T.H.-I. & Eastern Traction Co.		Water-tube	160		0	0		Cast-iron header failed
Feb. 16	Elkhart, Ind.	"Little Casino" Restaurant	3:30 a.m.	Cast-iron sectional	70	No. 20	0	1	$300	Unknown
Feb. 16	South Boston, Mass.	Northern Stores & Dock Corp., D St.		Cast-iron sectional	3		0	0	Repl'c'g sect.	Section cracked. Low water or flaw
Feb. 17	Schuylkill, Penn.	Losch Opera House					0	0		
Feb. 18	Lake Charles, La.	Hi-Mount Bakery	6:00 a.m.	Vertical	15	2 hp.	0	0	Boiler $20	Sacks piled on safety valve lever
Feb. 18	Pottstown, Penn.	Storb Trappe Rock Co.		Locomotive type			0	0		Crown sheet failed
Feb. 18	Independence, Kan.	Carl Leon Hotel		Cast-iron heating	10	M. 40-11	0	0		Two sections cracked. "Negligence"
Feb. 19	Baltimore, Md.	Continental Trust Co.		Water-tube			0	0		Cast-iron header failed
Feb. 19	Kalamazoo, Mich.	King Paper Co.		Water-tube			0	0		Tube failed
Feb. 21	Milwaukee, Wis.	Mill. Coke & Gas Co.		Water-tube	150	400 hp.	0	0		Six cast-iron headers failed
Feb. 21	Wheeling, W. Va.	Wheeling Steel & Iron Co.		Water-tube	110		0	0	$75	Tube ruptured
Feb. 22	Chicago, Ill.	Swift & Co. (Bldg. 20)		Water-tube			0	0		Tube ruptured
Feb. 22	Winchester, Mass.	Residence J. C. Kelley	5:00 a.m.	Water heater	45	16"x5'	0	0		Check valve on supply and no relief
Feb. 22	Far Rockaway, L. I., N. Y.	Residence W. I. Wolfe					0	0	$4000	
Feb. 22	Bellevue, Ohio	Bellevue Light & Power Co.	2:30 p.m.	Return-tube	100	100 hp.	2	0		Undue pressure. Flaw in belly sheet
Feb. 23	Philadelphia, Penn.	Miller Lock Co.		Water-tube			0	0		Tube failure

BOILER EXPLOSIONS DURING FIRST HALF OF 1914—Continued

Date	Place	Plant	Time of Day	Kind of Boiler	Pressure Carried, Lb.	Size	Killed	Injured	Property Loss	Probable Cause
Feb. 24	Philadelphia, Penn.	Mansbach & Co.					0	0		
Feb. 24	Boston, Mass.	"The Sheffield" Apartment House	11:30 p.m.	Cast-iron sectional	15	12 sections	0	0	Repairs	Low water
Feb. 24	Troy, Ohio	City Pumping Station	3:00 p.m.	Water-tube	150	100 hp.	0	0	$80	Cast-iron header failed
Feb. 24	Indianapolis, Ind.	American Hominy Co.		Locomotive boiler			0	2	$3000	
Feb. 24	Peabody, Mass.	National Calfskin Co. (Tannery)					0	0		Four tubes collapsed
Feb. 24	Kansas City, Kan.	Peet Bros. Mfg. Co.		Water-tube			0	1		Tube failure
Feb. 25	Hartford, Conn.	Sisters Holy Ghost Corp.		Cast-iron sectional			0	0		Sections cracked
Feb. 25	New York City, N. Y.	Apartment House, 111 E. 113 St.		Cast-iron sectional			0	0		
Feb. 25	Dayton, Ohio	Barney & Smith Car Co.		Water-tube			0	1		Tube ruptured
Feb. 26	New York, N. Y.	Apartment House No. 1757 3d Ave.		Hot-water heater			0	3		
Feb. 26	Chicago, Ill.	Crane Co.		Water-tube			0	0		Tube ruptured
Feb. 26	St. Joseph, Mo.	High School, St. Joseph					0	1		Blow-off failed
Feb. 27	Newburgh, N. Y.	Grace M. E. Church					0	0		
Feb. 27	Indianapolis, Ind.	Chapman Steel Co.	3:00 p.m.	Return-tubular		130 hp.	0	0	Cost patch	Scale caused bag. No explosion
Feb. 27	Kennett, Mo.	Willis & Lasswell's Dredge Boat	9:30 p.m.	Locomotive type	100	60"x14'	0	4	$5000	Low water
Feb. 27	Birmingham, Ala.	Birmingham Infirmary		Cast-iron sectional	10		0	0	$300	Sections cracked
Feb. 27	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	150 hp.	0	0	$50	Cast-iron header failed
Feb. 28	Sigourney, Ia.	Sigourney Electric Co.	9:30 p.m.	Return-tubular	135	200 hp.	0	1		Blow-off failed
Mar. 2	Canton, N. C.	Restaurant	5:00 a.m.	Water back in range	50		0	1		Pipes frozen
Mar. 2	Des Moines, Ia.	St. John's Lutheran Church	11:00 p.m.	Cast-iron heating			0	0	$12,000	Excess pressure
Mar. 2	Baltimore, Md.	Wm. A. Leonard's Book Bindery		Hotwater heater			0	2		
Mar. 2	Rockford, Ohio	Rockford Electric Light Co.	9:00 p.m.	Scotch marine	135	8'x13'	0	1	$1400	Furnace collapsed
Mar. 3	Indianapolis, Ind.	Aetna Laundry Co.	9:00 a.m.	Marine type	80	40 hp.	0	0	$400	Unknown
Mar. 3	Stephen, Minn.	Electric Light & Water Works					0	0		
Mar. 3	Howard City, Mich.	Fred Le Barr's Farm	4:00 p.m.	Locomotive type	100	10 hp.	0	2	Boiler only	Old boiler, excessive pressure
Mar. 3	Terre Haute, Ind.	T. H. Ind. & Eastern Traction Co.		Water-tube	160		0	0		Cast-iron header failed
Mar. 4	Sharon, Penn.	Carnegie Steel Co., So. Sharon		Water-tube			0	1		Tube failure
Mar. 4	St. Louis, Mo.	5670 Clemens Ave.					0	1		
Mar. 5	Kansas City, Mo.	Irish Cleaning Shop					0	0	$1000	
Mar. 5	Detroit, Mich.	R. H. Cook's Home		Cast-iron heating			0	0	$300	Sections failed
Mar. 6	Terre Haute, Ind.	Indiana Milling Co.					0	0		
Mar. 6	Syracuse, N. Y.	Sanderson Steel Plant					1	0		
Mar. 7	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	15	0	0		Cast-iron header failed
Mar. 7	N. Y. City, N. Y.	Carrol Box & Lumber Co., E. 18 St.					0	5		Blow-off failed
Mar. 7	St. Louis, Mo.	Bueter Hawkins Specialty Co.					0	1		
Mar. 8	Morganza, Penn.	Training School, Commonwealth Pa.		Cast-iron sectional			0	0		Sections failed
Mar. 8	Wyandotte, Mich.	Michigan Alkali Co.		Water-tube			0	0		Tube failed
Mar. 9	Waynesboro, Penn.	Landis Machine Co.	8:00 a.m.	Return-tubular	20	96"x20'	0	0	$40	Low water gage obstructed
Mar. 9	Berwick, Pa.	Chas. Shaffer's Home	11:00 a.m.	Cast-iron sectional	3	for 11 rooms	0	0	$75	Unknown
Mar. 10	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	150	0	0		Cast-iron header failed
Mar. 10	Benford, Tex.	Benford Lumber & Mfg. Co.					2	3		
Mar. 10	Burlington, N. J.	Albert Darr's Box Factory					0	2		Attempting repair under pressure
Mar. 12	Wellsville, N. Y.	Griswold Bros. (Fulner Valley)					0	1		
Mar. 13	New London, Conn.	N. Y. N. H. & H. R.R. Locomotive	10:13 p.m.	A.L.W. radial stayed			3	0	$400	Low water, crown sheet failed
Mar. 14	Buffalo, N. Y.	Central Casket Mfg. Co.		Cast-iron heating			0	0		Sections failed
Mar. 16	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	150	0	0		Cast-iron header failed
Mar. 16	Wallace, Idaho	City Dye Works	10:00 a.m.				0	0	$1500	
Mar. 16	Kalamazoo, Mich	Wm. Shakespere, Jr., Plant					0	1		
Mar. 17	Madisonville, Ky	Ruby Lumber Co.	7:00 a.m.	Return tubular	100	60"x16"	1	2	$100	Blow-off failed
Mar. 17	Troy, Ohio	City Pumping Station	1:30 p.m.	Water-tube		100 hp.	0	0	$200	Two cast-iron headers failed
Mar. 18	Charlotte, N. C.	Central Hotel		Cast-iron heating			0	0		Sections failed
Mar. 18	Brodrick, Penn.	John Antonia, Saloon	2:00 a.m.	Cast-iron heating	20	"426 S"	0	0	$5000	Unknown
Mar. 18	Jackson, Mich.	City Hall Bldg.					0	0		Flues burst
Mar. 18	New Canaan, Conn.	Southern New England Tel. Co.		Cast-iron heating			0	0		Sections failed
Mar. 18	Pen Argyle, Penn.	Penn Bangor State Quarry	2:20 a.m.	Return-tubular	110	125 hp.	1	0	$7300	Not known
Mar. 19	Grassland, Ill.	Pumping Station Southern R.R.	1:30 p.m.				1	0		
Mar. 19	Tiltonville, Ohio	Prime, Western Smelting Co.	7:00 a.m.	Hoisting engine			3	3		Over pressure
Mar. 20	Lake City, Mich.	F. H. Nye's Sawmill	7:30 a.m.	Locomotive type	120	10 hp.	1	1	$150	Safety valve (above roof) frozen, old boiler
Mar. 21	N. Y. City, N. Y.	Apartment House, E. 42 St.		Cast-iron heating			0	0		Sections failed
Mar. 21	Tolluca, Ill.	C. & A. Locomotive					0	0		
Mar. 21	Clarksburgh, W. Va.	Box Factory	10:00 a.m.	Return tubular	90	150 hp.	1	3		Blow-off pipe failure
Mar. 23	Easton, Penn.	Richter's Store and House					0	0		
Mar. 23	Bowling Green, Ky.	W. T. Smith, Sawmill	4:00 p.m.				0	1	$3000	Flaw in boiler
Mar. 26	Crowley, La.	La. & Western C.H.& S.A. R.R.	2:20 a.m.	Locomotive			0	2	$4500	Low water
Mar. 27	Carnegie, Penn.	Superior Steel Co.		Water-tube			0	1		Tube ruptured
Mar. 28	Medina, N. Y.	Maryland Dredging & Const. Co.		Locomotive type			0	0		Crown sheet failed
Mar. 28	Mishawaka, Ind.	Dodge Mfg. Co.		Water-tube	125	150	0	0		Cast-iron header failed
Mar. 31	Memphis, Tenn.	Memphis Steam Laundry Co.	11:00 a.m.	Water-tube	120	350 hp.	0	0		Cast-iron headers failed; 7 in front and 2 in rear
Mar. 31	Fanwood, N. J.	P. & R. Ry. Locomotive	4:30 a.m.	Locomotive type			0	2		Low water
Apr. 2	Boston, Mass.	"The Sheffield" Apt.		Cast-iron sectional			0	0		Four sections ruptured
Apr. 2	Boston, Mass. (Olive St.).	W. H. Gallison Co.					0	0		
Apr. 2	Boston, Mass., No. 394 Mass. Ave.	"The Sheffield" Apt. (2 same day)	11:30 p.m.	Cast-iron sectional	15	12 section	0	0	$460	Low water, neglect
Apr. 2	Westville, N. S.	Drummond Collieries	9:00 a.m.	Water-Tube	100		7	5		
Apr. 2	Monmouth, Ill.	Monmouth College	3:00 p.m.	Return-tubular	50	60"x16'	0	0	$50	Blow-off failed by sudden closing of valve
Apr. 2	Rockwood, Mich.	American Silica Co.		Return-tubular		125 hp.	0	1		Blow-off pipe failed

BOILER EXPLOSIONS DURING FIRST HALF OF 1914—Continued

Date	Place	Plant	Time of Day	Kind of Boiler	Pressure Carried, Lb.	Size	Killed	Injured	Property Loss	Probable Cause
Apr. 2	Monmouth, Ill.	Monmouth College	9:00 p.m.	Return-tubular		125 hp.	0	1		Blow-off pipe failed
Apr. 2	Rockwood, Mich.	American Silica Co.								
Apr. 3	Seattle, Wash.	Schwarger-Nettleton Mills		Water-tube			0	0		Tube ruptured
Apr. 3	Cedarhurst, L. I.	G. J. Waters (Residence)					0	0	$6000	
Apr. 4	Richmond, Va.	W. J. Parrish's Brickyard					0	0		Oil in boiler caused overheating
Apr. 6	Norfolk, Va.	Torpedo Boat "Aylwin"	4:20 a.m.	White-Forster (naval)	250	4000 hp.	3	0		
Apr. 7	Richmond, Va.	W. J. Parrish's Brickyard					0	0		Oil in boiler caused overheating
Apr. 10	Grand Rapids, Mich.	La Cross Gas & El. Co. Plant		Water-tube			0	0		Tube ruptured
Apr. 10	Pensacola, Leon Co., Fla.	Ed. Manning, owner, Sawmill	2:00 p.m.	Locomotive type	110		1	0		Low water
Apr. 13	Parishville Center, N. Y.	Sawmill	9:00 a.m.	Locomotive type	125	18 hp.	0	1	$50	Old age, mud and overpressure
Apr. 13	Woodward, Ala.	Woodward Iron Co.					0	2		
Apr. 13	Starkville, Miss.	City El. & Water Works		Water-tube			0	0		
Apr. 13	Cannonsburgh, Penn.	West Penn. Oil Co.	10:00 a.m.				0	0		
Apr. 14	Lutherville, Ark.	Read's Sawmill	9:30 a.m.	Locomotive type	70	40 hp.	1	0	$200	Low water
Apr. 15	Omaha, Neb.	Swift & Co.		Water-tube			0	1		Tube ruptured
Apr. 15	South Amboy, N. J.	Tugboat "Arles"					0	2		
Apr. 15	Omaha, Neb.	Burgess-Nash Co.'s Store	10:00 a.m.	Water-tube	135	150 hp.	0	1	$150	Tube ruptured
Apr. 16	Lebanon, Ind.	Lebanon Mfg. Co.					2	0		
Apr. 16	Estabouchie, Miss.	Rush Mill	6:00 p.m.	Return-tubular	100		0	0	$500	Low water
Apr. 16	Cannonsburgh, Penn.	Sawmill, McConnells Farm					0	0		
Apr. 17	Glassport, Penn.	American Axe & Tool Co.	7:00 a.m.	Water-tube		250 hp.	0	0	$100	Tube ruptured; defective tube
Apr. 21	Bangor, Me.	Orono Pulp & Paper Co.		Water-tube			0	0		Tube ruptured
Apr. 21	Brooklyn, N. Y.	Israel Rokeach					0	3		Blow-off failed
Apr. 22	Brooklyn, N. Y.	Hamilton Club		Water-tube			0	1		Tube failed
Apr. 22	Lindale, Ga.	Mass. Mills in Ga.		Water-tube			0	0		Tube ruptured
Apr. 23	Chicago, Ill.	Armour & Co., Glycerine Works		Water-tube			0	0		Two tubes ruptured
Apr. 23	Hartford, Conn.	Boskwick's Sheet Metal Plant		Cast-iron heating			0	0		Two sections ruptured
Apr. 25	Omaha, Neb.	Peoples Ice & Cold S. Co.		Water-tube	125	250 hp.	0	1	$50	Tube ruptured, faulty tube
Apr. 26	Omaha, Neb.	Swift & Co.,		Water-tube			0	0		Tube ruptured
Apr. 29	Allentown, Penn.	City Water-Works					0	0	$600	
May 1	Duluth, Minn.	M., St. P., Sault Ste. Marie Ry. Co.					0	0		Blow-off failed
May 1	Jersey City	Chas. Lehman		Laundry			0	0		
May 1	Pickwick, La.	Pickwick Lumber Co.					2	0		
May 1	Columbia, Miss.	Enterprise Lumber Co. Sawmill					1	4		
May 2	Ada, Okla.	Oklahoma Portland Cement Co.	4:00 a.m.	Water-tube		250 hp.	0	0	$2.50	Tube ruptured
May 4	Arden, near Grafton, W. Va.	B. & O. Locomotive	8:10 a.m.	Locomotive	200		1	1		Low water, crown sheet failed
May 5	Blackville, S. C.	Blackville Lighting Co.					0	1		
May 8	Wilmington, N. C.	Government Str. "Mercur"					0	2		
May 9	New Baltimore, Mich.	Shore Line Railway Power House	5:15 p.m.	Water-tube		300 hp.	2	1		Tubes pulled out of new steel muddrum
May 9	Pittsburg, Kan.	Joplin & Pittsburgh Ry. Co.					0	0		Tube ruptured
May 10	Altoona, Penn.	Standard Ice Co.		Steam pipe			0	1		
May 10	Bradentown, Fla.	Excelsior Ice Co.	11:30 p.m.	Return-tubular		50 hp.	0	0	$64	Scale
May 11	Norfolk, Va.	S.S. "Jefferson," O.D. Line	10:55 p.m.	Scotch marine	165		11	0		Tube sheet
May 11	Franklyn, Penn.	Public School		Cast-iron heating			0	0		Sections failed
May 12	Waterbury, Conn.	Scoville Mfg. Co.		Water-tube			0	0		Tube ruptured
May 13	Lebanon, Penn.	Semet-Solvay Co.		Water-tube			0	0		Tube ruptured
May 13	Renfrew, Penn.	Pittsburgh & Butler Ry. Co.		Water-tube			0	0		Tube ruptured
May 13	Detroit, Mich.	Detroit Copper & Brass Co.	1:15 p.m.	Water-tube	125	94" drum	1	0	Nominal	Loose scale obstructed tube, tube failure
May 13	Canal Dover, Ohio	Hardestry Milling Co.					0	1		Flue burst
May 14	Hoboken, N. J.	Hoboken Paper Mill					0	1		
May 14	Parishville Center, N. Y.	Stone Sawmill					0	0		
May 14	Waterbury, Conn.	Scoville Mfg. Co.		Water-tube			0	0		Tube ruptured
May 15	Security, Md.	Hagerstown & Frederick Ry. Co.	11:00 a.m.	Water-tube		700 hp.	0	0		Tube ruptured
May 18	Madison, Wis.	Hauseman Brewing Co.		Water-tube			0	0		Tube ruptured
May 18	Scranton, Penn.	Undergood Colliery					0	1		Steam pipe burst
May 19	Bradentown, Fla.	Excelsior Ice Co.	12 noon	Return-tubular		50 hp.	0	0	$35	Scale
May 20	Ecorse, Mich.	Worcester Salt Co.					0	0		
May 21	Vinal Haven, Me.	Bodwell Granite Co.					0	0		Furnace failed
May 23	San Francisco, Calif.	150 Erie St.					0	1		
May 24	Waterbury, Conn.	Scoville Mfg. Co.		Water-tube			0	0		Tube ruptured
May 25	Greensboro, N. C.	Yow's Laundry		Range boiler			1	2	$900	Check in supply, safety valve inoperative
May 27	Sault Ste. Marie, Mich.	Dudo Bros. Sand & Mill Co.					0	0		
May 27	Bellows, Vt.	Moore's Paper Mill	7:00 a.m.	Return-tubular	90	6x18'	1	0		Valve on steam pipe burst
May 27	Stearns, Ky.	William's Mill	10:00 a.m.	Portable	120	30 hp.	0	4	Boiler	Low water
May 27	Newark & Harrison, N. J.	Standard Engineering Co., Dredge					1	3		Tube blew out
May 29	Joliet, Ill.	Fredrick Cowan Co.		Water-tube			0	0		Tube ruptured
May 29	Easley, S. C.	Easley Cotton Mills		Water-tube			0	0		Tubes ruptured
May 29	Seekonk, Mass.	H. E. Cushing's Farm	10:00 a.m.	Vertical	80	15 hp.	1	2		Old boiler, not inspected
May 29	Ten Mile Lake, Ore.	Muetzel Sawmill					2	3	$200	Over pressure
May 30	Brooklyn, N. Y.	Foster Pump Works					0	1		
May 30	Brooklyn, N. Y.	984 Franklyn Ave.	11:00 a.m.	Water-tube	75	150 hp.	0	1		Two tubes ruptured
May 30	Carfile, Ark.	Loutre Shingle Co.	5:00 a.m.	Locomotive			0	0	$800	Crown sheet failed
May 30	Oakdale, La.	Bouman Hicks Lumber Co.	2:35 p.m.	Locomotive	145		5	1	$3500	Broken staybolt
June 2	Puerto, Mexico	Cruiser "Salem"		Water-tube			1	2		Tube burst
June 2	Salamanca, N. Y.	Tunnessa Lumber Co., Locomotive		Air tank			1	0		
June 3	Beaverton, Ore.	Cedar Mills Rock Quarry	4:00 p.m.	Locomotive type	95	13 hp.	0	1		Crown sheet failed
June 3	Fibre, Mich.	Dudo Bros. Mill					0	0	$2500	Safety valve failed
June 3	Duvall, near Columbus, Ohio	Locomotive, N. & W. R.R.	3:45 a.m.	Locomotive	180		2	0	$20,000	Low water, crown sheet failed
June 5	Crookston, Minn.	Fredricks Bros.' Farm		Locomotive type		20 hp.	1	1	$1000	Over pressure
June 6	Racine, Wis.	A. J. Horlick		Hot-water boiler			0	0		

BOILER EXPLOSIONS DURING FIRST HALF OF 1914—Continued

Date	Place	Plant	Time of Day	Kind of Boiler	Pressure Carried, Lb.	Size	Killed	Injured	Property Loss	Probable Cause
June 8	Washington, D.C.	Heurich Brewery		Water-tube	150		0	1		Tube failure
June 8	New York, N. Y.	Apt. House No. 137 E. 33d St.		Hot-water boiler			0	0		
June 8	Trenton, N. J.	"Twilight", D. R. Trans. Co.					0	0		"Valve blown out"
June 8	Stamford, Conn.	Locomotive	5:30 a.m.	Locomotive			0	2		Steam pipe blew out
June 8	Meridian, Miss.	Meridian Light & Railway Co.	6:45 a.m.	Water-tube	150	250 hp.	1	2	$35,000	"Breathing" of the drum head
June 8	Leonard, Tex.	Leonard Light & Ice Co.					0	0		
June 9	Monongahela, Penn.	Diamond Coal Co. (Pump Boat)	1:30 a.m.				0	1		
June 10	Davenport, Iowa	Crescent Macaroni & Cracker Co.	6:30 a.m.	Return-tubular	110	60"x18	0	0		Low water
June 11	Philadelphia, Penn.	Bonfig Bakery					0	1		
June 12	Toronto, Ont.	Bowes Produce Co.		Water-tube			0	3		Pipe bursted
June 12	Battle Creek, Mich.	Michigan Carton Co.		Water heater		1000 b.hp.	0	0		Tube ruptured
June 13	Moline, Ill.	Deere Harvester Co.	8:00 p.m.	Return-tubular		60 hp.	0	1		Back pressure valve stuck
June 13	Richmond, Va.	Millhiser Bag Co.		Locomotive type	100	15 hp.	0	0	$125	Weak seam indicated; no explosion
June 13	Franklyn, Ga.	J. P. Pyles Farm	10:00 a.m.	Hot-water heater			4	5	$200	Low water, old boiler
June 15	Franklin, N. H.	Ideal Hotel					0	0		
June 15	Anderson, Ga.	Tyus' Soda Water Stand					0	1		Charging tank burst
June 17	Oakland, Calif.	Piedmont Baths					1	1		
June 20	Milledgeville, Ga.	Milledgeville Ice Co.					2	4		
June 21	Charleston, S. C.	Charleston Cons. Ry. & Lt. Co.		Water-tube			0	0		Section cracked
June 22	Jennings Gap, Va.	Geo. Shull's Home					0	1		Low water, crown sheet failed
June 22	Laird, Kan.	C. W. Smith Elect. & Ice. Co.					0	0		
June 22	Oscoda, Mich.	Portable Sawmill					1	0		Tube ruptured
June 22	Bosler, Wyo.	Portable Sawmill					1	0		
June 22	Toledo, Ohio	St. Michaels Church		Cast-iron sectional			0	0		Blow-off failed
June 22	Arthur (Roanoke) Va.	Norfolk & Western Locomotive	6:15 p.m.	Locomotive	200	$8000	2	0		Section cracked
June 22	Peekskill, N. Y.	Peekskill Hat Mfg. Co.					0	0		Low water, crown sheet failed
June 23	Jennings Gap, Va.	Feed Mill	1:00 p.m.	Locomotive type		12 hp.	0	1		
June 23	Littleton, Ill.	J. C. Decker Novelty Works					0	1	$100	Low water
June 24	Williamsport, Penn.	Brown Dredging Co.					0	2		
June 25	Evansville, Ind.	J. W. McCutcheon (King St.)	9:00 a.m.	Portable well drill	80	20 hp.	1	1	$1000	Corrosion at bottom of fire-box.
June 25	Oshawa, Can.	I. L. Prew's Machine Shop					0	0		Blow-off failed
June 26	Fall River, Mass.	Desplaines Safety Deposit Co.		Water-tube			0	1		Tube ruptured
June 26	North Bend, Ore.	Muetzel Sawmill	11:00 a.m.				2	3	$1000	Low water
June 26	Chicago, Ill.	A. J. Johnson & Sons Furniture Co.					0	1		Blow-off failed
June 27	Mattoon, Ill.	London Packing Co.		Water-tube			0	0		Tube ruptured
June 27	Philadelphia, Penn.	Standard Roller Bearing	7:30 p.m.	Return-tubular		150 hp.	0	1	$1.00	Blow-off failed

Water-Power Legislation

In the deadlock between the President and the Senate over patronage, waterpower legislation at this session of Congress seems likely to be lost in the shuffle. States-right senators, like Smoot of Utah, Borah of Idaho and Clarke of Wyoming, backed by big water-power interests, are understood to be preparing to talk to death the Ferris bill, which has passed the House, providing for the leasing of power sites in the public domain, while the conservationists have declared war on the Shields bill, reported to the Senate as a substitute for the Adamson general dam bill which passed the House last year. The Shields bill is denounced by Gifford Pinchot, president of the National Conservation Association, as "the bill of the water-power monopolists."

The Adamson bill, which had the indorsement of President Wilson, provided for the granting of permits for dams and power plants on navigable streams for periods of not more than fifty years, with reversion of the property to the Government at the end of that period. The Shields bill, while it purports to do this same thing, leaves the way open to endless litigation for determining the fair value of the property at the end of the franchise term, so that it is declared by conservationists that it would be practically impossible for the Government to ever take over the property. Although the only grant to be made by the Government is for the right to build a dam and works adjacent to a navigable stream, the language of the Shields bill would require the United States to take over the entire lighting plant of a city, if it were operated in connection with such power plant, in order to recapture the water power in the stream. The Shields bill also gives to the water-power interests the right to condemn land, either public or private, for their own uses, and would require the Government, on taking over the plant, to pay the increased value of land so taken.

These features of the bill, which are not the only ones to which the conservationists object, are in direct opposition to the policies declared for by Secretary Lane, of the Interior Department, and at odds with the principles set forth in the Ferris bill for the granting of franchises for power sites on public lands.

Of the Shields bill, Mr. Pinchot says in his statement:

There has been no clearer attempt to defeat the conservation policy since water power first became a great National problem. It is a direct reversal of the wise and fair provisions contained in the Adamson bill as it passed the House. President Roosevelt vetoed the James River bill in 1909, President Taft vetoed the Coosa River Dam bill in 1911, because they did not provide for proper payment to the public for value received from the power companies. The House of Representatives by an overwhelming vote adopted the Sherley amendment providing for such payment. The Shields bill proposes to give these rights away. It is a surrender to the special interests, and its passage would be a public calamity.

Recent Court Decisions

Digested by A. L. H. STREET

Remedy for Breach of Power Contract—According to a decision lately handed down by the Georgia Supreme Court, an electric-power company which has an unexpired exclusive contract to furnish a consumer with all electricity used by him for power purposes cannot maintain a suit to enjoin breach of the contract through the consumer's refusal to receive further service, where it appears that the company has an adequate remedy by suing for damages for such breach. (Macon Ry. & Light Co. vs. Palace Amusement Co., 83 "Southeastern Reporter," 105.)

Power as Basis of Mechanic's Lien—The Minnesota Supreme Court has just been called upon to determine the interesting question whether one who furnishes fuel and lubricating oils for the production of power in operating excavating machinery in constructing a building foundation is entitled to a lien against the real estate to secure payment for the fuel and oils on the theory of having contributed toward the improvement of the land. (Johnson vs. Starret, 149 "Northwestern Reporter," 6.) In answering this question in the affirmative, the court said: "Had the excavation and removal of the earth been done by manual labor, the right to a lien therefor would be undoubted, and we cannot differentiate such a case from one where the same result is reached by other and modern methods. The value of the defendant's property was thereby enhanced, and it can make no difference that this was accomplished by the use of power obtained from materials furnished by the lien claimants instead of by common labor."

Thermodynamics of the Marine Oil Engine*

By John F. Wentworth

SYNOPSIS—This paper is intended to present certain experimental data obtained by the writer as well as to reintroduce certain old and well established facts in a new garb, in order that the oil engine may be looked at from a new viewpoint.

The marine plant must be capable of running at full load with maximum power or of running at partial load with maximum efficiency, and most essential at times is the ability to run at extremely slow speeds. Moreover, to meet all conditions and take its place as a perfect prime mover at sea the marine oil engine must be capable of being built in large single units, which means that extreme pressures must be avoided. Therefore, an attempt will be made to show:

First, that if possible the extreme high pressure of compression must be reduced.

Second, that to give the greatest possible efficiency under all conditions the proportions of air to fuel must be kept constant regardless of load.

Third, that to get extreme or emergency slow speed the injection of the fuel and the ratio of air to fuel should be varied contrary to the condition for maximum efficiency.

Fourth, that the percentage of the stroke during which the fuel valve is held open should be under the control of the operator in order that injection air losses may be reduced.

Fifth, that by reducing the compression and at the same time hastening the fuel injection the advantage of the high compression is not materially impaired.

Sixth, that by reducing the compression it is possible to obtain substantially the same power with the same theoretical efficiency, but with an increased mechanical efficiency.

Seventh, that by increasing the temperature of the injection air a saving of practically 10 per cent. of the fuel now used may be effected without danger to the plant.

To understand best the effect of the high-pressure of compression and its relation to the cycle, temperature-volume diagrams of the oil engine and of other types will be considered. Fig. 1 shows two typical Diesel diagrams copied from catalogs of 1898 and 1913. The end of the expansion stroke corresponds with 100 per cent. volume and the beginning with the clearance of the engine, making it possible to read pressures and also percentage volume. Assuming that the compression is begun with air at 14.7 lb. and 60 deg. F., the specific volume will be 13.09, and the specific volume at any other point will be 13.09 times the percentage volume. Thus, from the formula $PV = RT$, the temperature of the charge can be figured with a fair degree of accuracy. The constant R for air is 53.22 in English units. By plotting the two indicator diagrams and figuring a certain number of points the temperature-volume diagrams were drawn in. It will be noted first that the temperature rises almost vertically during injection. This shows, comparing the 1898 and the 1913 diagrams, that the trend of Diesel engine practice approaches the conditions of the gas engine.

If the Diesel engine can start with a compression of 500 lb., the charge igniting at a temperature around 920 deg. F., then after it has been running awhile the temperature at the end of compression must be around 1450 deg. F. If this be so, the charge will ignite at any time after the pressure has exceeded 120 lb., provided the engine has run long enough to become normally warm.

This is demonstrated by the diagrams shown in Fig. 2, which were taken from the writer's experimental engine, although it was not possible to vary the amount of air compressed per stroke as much as desired. Also, the timing of the fuel valve caused slight trouble. Hence, the fuel valve was arranged so that the timing of the injection could be changed, and these diagrams are the result. So far as the writer knows, this is the first instance in which a timing arrangement has been used on the fuel valve. The ignition of the fuel was obtained under conditions which were possible only in an engine which had been run long enough to get warmed up, and it seems a fair assumption that, if the fuel ignited as shown, it would have ignited if the clearance had been so increased that these low pressures were obtained at the end of compression. The fuel is shown igniting at from 70 lb. up to full compression pressure. Diagram No. 6, where the ignition was at 130 lb., makes it apparent that much energy is lost by radiation during the time of high pressure and high temperature. This loss at the end of compression should be somewhat reduced in the low-pressure type, for although the cylinder volume at the end of compression might be doubled, the radiating surface would be only slightly increased.

Next consider the proposition that the ratio of air to fuel be kept constant. This has been done on the best gas engines as a means of governing, but it has not been done on the Diesel, because apparently the full amount of air must be compressed per stroke in order that the ignition temperature may be obtained. In the Mar. 11, 1913, issue of "Power" the writer brought this out, proposing that if only two-thirds of the fuel were used, only two-thirds of the regular amount of air should be compressed. Moreover, if the friction loss is a function of the unit pressure on the piston, then the friction of the Diesel at two-thirds load would be 50 per cent. greater than in the proposed form of governing. This can be stated in another way; namely, the friction is not a percentage of the net load, but is a percentage of the work done in the cylinder, regardless of whether the effect is plus or minus on the brake.

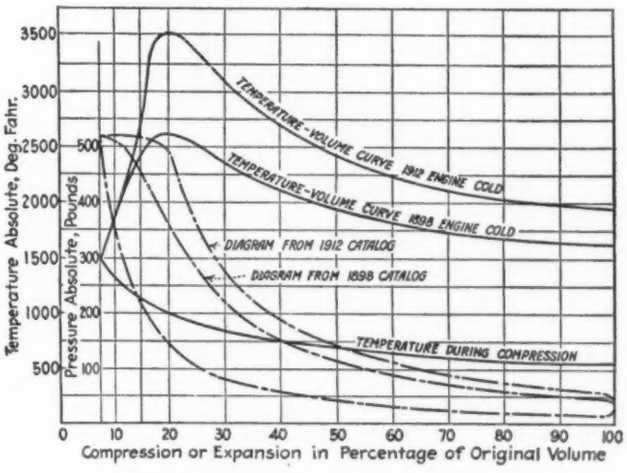


Fig. 1. Temperature-Volume Curves

Fig. 3 is a set of curves constructed from tests made by Professor Denton in 1898 on a small Diesel engine. Improvements have been made in the engine, but the cycle is unchanged, so that results obtained in these early tests are at least indicative of what goes on in the present engines. This diagram shows that the friction is practically constant at all loads and would seem to bear out the contention that the friction is toll taken out of both the compression and the working strokes. With the compression pressure constant the sum of the compression work and expansion work would vary but little for a wide variation in the net work done in the cylinder. However, the sum of the compression and the expansion strokes would vary greatly if the compression pressure was decreased.

Air is compressed and then expanded. The air in itself does nothing. All the work put into compression will be given up in expansion except the losses. If the quantity used per stroke is reduced, the losses per stroke will be diminished by this same amount. Unnecessary compression of air is an extravagance, therefore, for it is needless to compress more than is required for the proper combustion of the fuel.

Next, consider the problem of extremely slow speed. In order to reduce to a very slow speed care must be taken to obtain a maximum temperature at the end of the compression stroke. Diagram No. 6, Fig. 2, shows the effect of cooling

*Excerpts from a paper read at the recent meeting of the Society of Naval Architects and Marine Engineers, at New York.

at the end of compression. For moderate speeds with a falling horsepower the amount of air compressed per stroke can be varied. Under present conditions there is a limit to the speed of the oil engine, at which limit the ignition will be uncertain. To go beyond this in the present engines the air should be heated before it enters the cylinder and the temperature of the jacket water should be raised. It might even be wise in an emergency to cut out the circulating water entirely.

Need for this slow speed was painfully evident when the Diesel-engined ship "Christian X" fell in with a disabled ship in midocean. It was stated at the time that tow lines were repeatedly passed to the disabled ship only to part. Presumably the best that could be done would have been to keep starting and stopping the engines in the hope of gradually accelerating the tow to the point where the line would stand the lowest possible speed of the engine.

A steam vessel under the same conditions would have run her engine very slowly, just enough to give steerage way, until she had taken up the slack of the tow line, and then would have increased the revolutions gradually until the desired speed was obtained. This is manifestly impossible in the present oil engine. In the plant proposed at the end of this paper the motive power would be steam and the engine would start at a few turns per minute with steam from the economizer boilers.

At this point it may not be out of place to call attention to the fact that the cylinders of a Diesel engine should be kept as hot as possible, up to the point of encountering lubricating troubles. This is true whether extremely slow speed or maximum efficiency is required.

Passing to a discussion of the advantages of fuel valve control as to timing, etc., in the present oil engine the duration of the fuel injection does not take into account the amount of fuel to be fed nor the speed. If an engine be slowed down to half speed, considerably less fuel will be used than when running at full speed. Notwithstanding this, the fuel valve is open for the same length of time. This results in a waste of injection air, the thermodynamics of which will be taken up later. If the fuel be injected early in the stroke, a rise in pressure can be obtained which will increase the efficiency. If the fuel be fed into the cylinder a considerable time after the end of the compression stroke, it is possible to produce the same result that is obtained in the explosion engine by retarding the spark. Thus, it seems to be highly desirable to be able to control at will both the timing of the injection and the duration of the fuel feed.

The method of compensating the theoretical loss encountered in reducing the compression pressure next calls for consideration.

If the compression pressure is reduced, the fuel injection can be hastened and combustion completed earlier in the stroke. If high compression is used, then the fuel must be slowly injected, as is now done in the Diesel engine in order that extreme pressure may be avoided. If the fuel be injected rapidly and timed for the end of high compression, results are to be expected similar to those shown in diagram No. 8, Fig. 2.

High compression means small clearance and a relatively large number of expansions, which is desirable, but the full effect is lost through the slow fuel injection. Why this is so is shown in Fig. 4. Assume the indicator diagram to be divided into four similar ones by means of the adiabatic lines shown. Each may be considered separately. In the case of D the average clearance is 16.1 per cent. of the whole cylinder contents, hence it has only 6.2 expansions. Diagram A, on the other hand, has 11.34 expansions. If the fuel be injected rapidly into a cylinder whose clearance gives a compression pressure of 275 lb., then the number of expansions will be 8, or the same as the average of this Diesel diagram.

Attention will now be directed to some of the benefits to be derived from reduced compression. In the engineering press the idea of reducing the compression pressure and at the same time making up for this lower compression temperature by means of a higher temperature of the incoming air has been opposed, the objection being that the volumetric efficiency would be decreased; in other words, the power of the cylinder would be reduced, making the engine more bulky. The original idea was to save fuel by a process similar to compounding; that is, when only a partial load was called for, a partial cylinderful of air would be compressed. If this were done by closing the admission valve at the middle of the suction stroke, a vacuum would be formed in the cylinder during the rest of the stroke which would be maintained during the first half of the compression stroke. In this case volumetric efficiency will not suffer, whereas in the present method of governing the engine at partial loads the efficiency is handicapped by the presence of an unnecessary excess of air.

The economy possible in the injection air is the last point for discussion pertaining to the present type of engine.

From the best information available the injection air compressor must develop about 10 per cent. of the brake horsepower of the main engine. Of this 10 per cent., what care has been used to maintain an economical cycle? The air is compressed in two or more stages to 1000 lb., and then cooled to the original temperature or lower. The total energy in the air after it has been compressed to 1000 lb. and cooled to its original temperature is exactly what it was in the beginning. All that the work of compression has done is to make a part of this original energy available. Our natural conception of the energy of compressed air would lead to the consideration that the energy depended upon the pressure of the air; as a matter of fact, the determining factor is not pressure, but temperature. There are several simple proofs for this statement. One is: "Isoenergic lines are lines representing changes during which the intrinsic energy remains constant. It will be seen later that the isoenergic and isothermal lines for a gas are the same" (Peabody). When the air

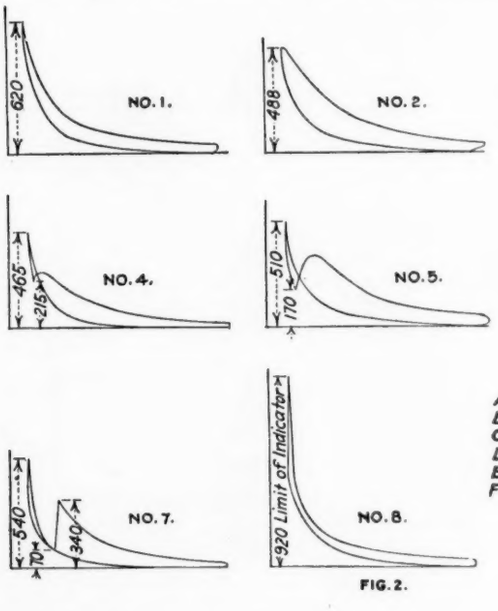


FIG. 2. DIAGRAMS TAKEN ON EXPERIMENTAL ENGINE

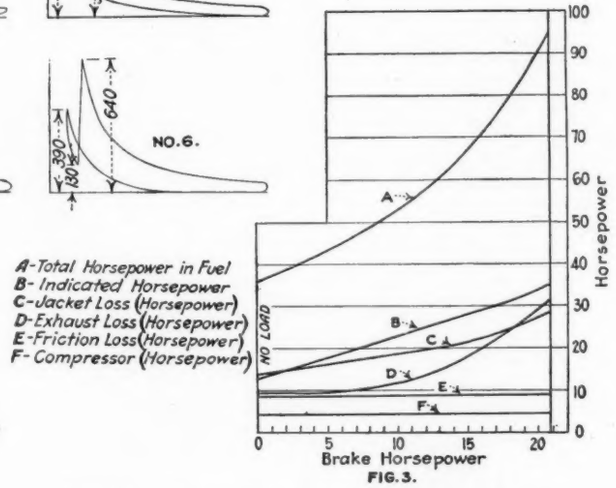


FIG. 3. VARIATION OF LOSSES WITH DIFFERENT LOADS

is cooled to the original temperature after having been compressed, it is brought back to the isothermal from which compression started.

This contention can be proved mathematically. The formula for work of expansion to the absolute zero of temperature and pressure is

$$W = PV \div (k - 1)$$

This formula considers a fixed quantity of air—that is, a certain weight. In the formula PV=RT, considering the same weight of air, P and V will be the same as in the formula for work. By substitution, the formula for work can be rewritten $W = RT \div (k - 1)$, where R and k are constants, and the only variable in the formula is T, the absolute temperature. This new formula will give the work of expansion for one pound of gas.

The above consideration of the injection air is simply to bring home the fact that a waste is being made of one-half of the available energy of the compressed air when this air is cooled to room temperature in place of being used at about 580 deg. F. If the injection air is used at this temperature, just one-half of the fuel to run the air compressor will be saved, and there will be no difference in the operation of the engine. Before this can be wisely done, however, there is need for redesigning of the fuel valve. It is the writer's opinion that the valve should be simplified even if the pump is complicated.

In regard to the danger in using air of this temperature there is this to say: Any explosion in the injection line will come from a quick opening of the air valve. If there is a pocket in the line where oil can settle and the air valve is suddenly opened with no pressure in the line, the body of oil will be dislodged from its pocket and shoved along ahead of the incoming high-pressure air, compressing the low-pressure

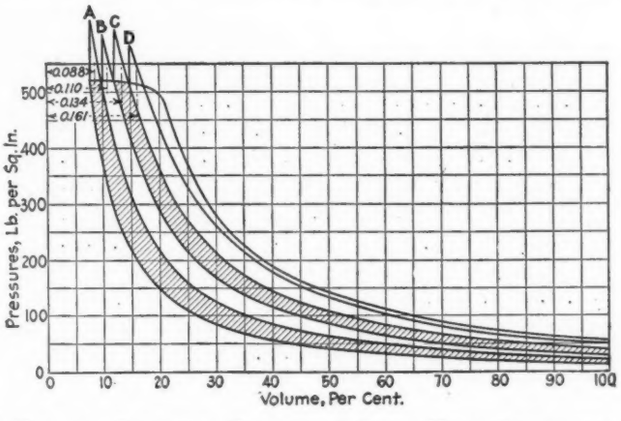


FIG. 4. SHOWING EFFECT OF SLOW FUEL INJECTION

air. If this action in the injection line takes place quickly, the body of oil will take the place of a piston and the action in the injection line will be the same as in the cylinder during the compression stroke. A temperature of ignition will be reached in the injection line regardless of the temperature of the injection air. In fact explosion is caused not by the injection air, but by the contents of the air line when the high-pressure air is turned on. It is possible to safeguard this type of engine from explosion in the injection line by a little care in the design of the piping arrangement, etc., and the exercise of due caution in operation.

The present instructions for the operation of Diesel engines state that the cooling water from the jacket of the fuel valve should be cold to the hand. The writer feels that the fuel valve can be designed so that it needs no jacket whatever. This may be objected to as an unsupported opinion, but it is based upon years of study along this line of work during which nothing has been found to oppose this idea and everything points to its possibility.

In closing, the writer's proposed thermo plant will be considered. Briefly stated, the starting medium is steam, generated in a special boiler. Before starting, the steam is admitted to the jackets of the engine, warming it to a point where starting steam can be used without excessive condensation. After the engine is running on oil the flow through the jackets is reversed, water from the bottom of the boiler enters the bottom of the jacket, and a mixture of steam and water comes from the top and is returned to the boiler. In this case combustion in the cylinder will not increase the temperature of the jacket water, but will transform it from water to steam at the same temperature. Also, the exhaust gases from the engine will pass around and through this boiler

TEMPERATURES AND PRESSURES CORRESPONDING TO DIFFERENT CRANK POSITIONS, TAKEN FROM TYPICAL DIAGRAMS

Degrees of Crank Angle		Compression												Combustion										
		180	195	210	225	240	255	270	285	300	315	330	345	360	375	390	405	420	435	450	465	480	495	510	525
Diesel engine (cold)	Temp. deg. F.	60	65	67	84	113	145	212	265	420	635	715	872	920	1170	1790	1840	2040	1740	1530	1325	1230	1170	1105	1090
	Press. lb. gage	0	0.3	1.1	2.3	5.3	9.0	17.0	30	57	121	222	410	505	501	438	270	190	122	80	56	44	36	32	30
Diesel engine (hot)	Temp. deg. F.	260	266	270	293	333	377	468	543	758	1055	1165	1381	1450	1790	2660	2720	2660	2580	2300	2010	1880	1800	1710	1690
	Press. lb. gage	0	0.3	1.1	2.3	5.3	9	17	30	57	121	222	410	505	501	438	270	190	122	80	56	44	36	32	30
Wentworth experimental engine (hot)	Temp. deg. F.	60	65	67	85	113	145	212	265	420	635	715	872	920	1570	1585	2000	1750	1440	1240	1140	990	940	915	900
	Press. lb. gage	—8.2	8.1	7.7	—7.2	—5.8	—4.1	—0.5	5.2	17.2	45	90	172	215	270	200	120	65	41	32	22	14	10	8	7
Gas engine (cold)	Temp. deg. F.	60	60	62	65	72	85	127	155	200	255	340	415	452	2500	2480	2380	2160	2050	2010	1850	1780	1660	1610	1507
	Press. lb. gage	0	0	0	1	1.4	3.1	5	10	13	24	39	80	90	305	260	190	143	111	81	70	60	45	35	20
Gas engine (hot)	Temp. deg. F.	260	260	265	268	277	295	325	364	426	531	647	752	802	3540	3520	3475	3200	3030	2960	2760	2640	2490	2410	2320
	Press. lb. gage	0	0	0	1	1.4	3.1	5	10	13	24	39	80	90	305	260	190	143	111	81	70	66	45	35	20
Wentworth experimental engine overload (cold)	Temp. deg. F.	60	64	68	75	85	106	140	210	290	452	627	790	918	1065	1940	2555	2680	2505	2370	2190	2200	2195	1955	1925
	Press. lb. gage	0	0.3	0.7	1	2.3	7.8	14	32	46	98	203	383	500	470	358	242	165	120	90	78	68	55	40	30

Degrees of Crank Angle		Exhaust												Suction										
		540	555	570	585	600	615	630	645	660	675	690	705	720	15	30	45	60	75	90	105	120	135	150	165
Diesel engine (cold)	Temp. deg. F.	600	450	450	450	450	450	450	450	450	450	450	450	450	60	60	60	60	60	60	60	60	60	60	60
	Press. lb. gage	15	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	0
Diesel engine (hot)	Temp. deg. F.	1465	988	988	988	988	988	988	988	988	988	988	988	988	600	550	400	300	280	270	260	260	260	260	260
	Press. lb. gage	15	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0.5	0	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	0
Wentworth experimental engine (hot)	Temp. deg. F.	890	890	890	890	890	890	890	890	890	890	890	890	890	700	600	400	300	212	145	135	84	67	65	60
	Press. lb. gage	0	0.2	0.2	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0	0.5	—0.5	—0.5	—0.5	—4.1	—5.8	—7.2	—7.7	—8.1	—8.2
Gas engine (cold)	Temp. deg. F.	1312	919	919	919	919	919	919	919	919	919	919	919	919	60	60	60	60	60	60	60	60	60	60	60
	Press. lb. gage	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0	0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	0
Gas engine (hot)	Temp. deg. F.	1960	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1450	1000	750	500	400	300	290	280	270	260	260	260
	Press. lb. gage	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	—0.5	0.0
Wentworth experimental engine overload (cold)	Temp. deg. F.	1560	1340	1160	1160	1160	1160	1160	1160	1160	1160	1160	1160	1160	60	60	60	60	60	60	60	60	60	60	60
	Press. lb. gage	15	5	0	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0	—0.3	—0.3	—0.3	—0.3	—0.3	—0.3	—0.3	—0.3	—0.3	—0.3	—0.3

before being rejected, and as much as possible of this waste heat will be converted into steam. The energy from the boiler will be used to run the auxiliaries and, if the idea is correct, to run a separate steam propelling plant. The exhaust from all steam units will be returned to a condenser, where at last the remaining heat units will be abandoned as unavailable energy.

OBITUARY

JOSEPH G. McCOLLUM

Joseph Grant McCollum, superintendent of construction of the Essex power station of the Public Service Electric Co., at Point-no-Point, on the Passaic River, died from pneumonia at the Newark (N. J.) Private Hospital, Jan. 13, at the age of 29.

He was graduated from Cornell University in the class of 1909. He was for a time with Westinghouse Church Kerr & Co. in New York, and early in 1914 removed to Newark, N. J., and became superintendent of construction at Burlington for the Public Service Electric Co. of New Jersey.

F. W. JENKINS

On Thursday, Jan. 14, 1915, Frank William Jenkins died at his home in Brooklyn, N. Y., from complications due to old age. He was widely known as an expert in hydraulics and was connected with the Henry R. Worthington Pump Co. for over fifty years.

Mr. Jenkins was born in Hudson, N. Y., Feb. 26, 1832, moved to Brooklyn at the age of 14, and spent the remainder of his life there. Many inventions and improvements in steam pumps and hydraulic engineering are the products of his genius. He also occupied a high place in the civic and municipal life of the comunity in which he lived. Two daughters survive him.

PERSONALS

William Naylor, after a service of 44 years in the engineering department of Marshall Field & Co., has retired from his position as chief engineer. He was born in Lancashire, England, Jan. 4, 1833, was apprenticed at the age of 9 to the London & North Western R.R. shops at Leeds, and was promoted to engine runner at the age of 19. In 1859 he was driving the "Manchester-Liverpool Flyer," which occupation he left to come to America, arriving at New Orleans after a voyage of seven weeks. He worked at New Orleans, Jackson and Memphis for short periods and in 1860 settled in Mt. Carmel, Ill., and engaged in the lumber-sawing business. He moved to Warrensburg, Mo., in 1865, and in 1866 to Chicago, where he entered the employ of Field, Leiter & Co., now Marshall Field & Co., in 1871. He is father of Past-President Chas. Naylor, of the N. A. S. E., of which organization he is himself an active member, being treasurer of the Robert Fulton Association No. 28, of Illinois, and is known to, and esteemed by, almost everyone in Chicago who is in any way connected with power-plant engineering.

NEW PUBLICATIONS

OIL FUEL, ITS SUPPLY, COMPOSITION AND APPLICATION. By Edward Butler. Published by Charles Griffin & Co., Ltd., London, and J. B. Lippincott Co., Philadelphia, 1914. Size 5x7½ in.; 328 pages; illustrated.

After introductory chapters on the origin, production and economic aspects of oil fuel, the author reviews some of the early work with oil fuel, compares the steam-, air- and pressure-jet methods and then proceeds to descriptions of the burners in use today for steam boiler furnaces. Oil fuel for marine and locomotive purposes is dealt with at length and its use in metallurgical work is also considered. Very wisely, it would seem, no attempt has been made to touch upon the internal combustion engine, as this subject is so broad in itself that any treatment in a book of this kind would necessarily be incomplete.

THE ANALYSIS OF COAL WITH PHENOL AS A SOLVENT. By S. W. Parr and H. F. Hadley, Bulletin No. 76 of the University of Illinois. Paper; size, 6x9 in.; 41 pages, illustrated. Price, 25c.

As far back as 1851, experiments were made on coal for the purpose of dissolving those constituents of the coal which were soluble in certain chemicals, and from time to time different investigators have taken up the problem. The experiments by the authors of this bulletin have been made for the purpose of overcoming some of the objections to both the chemical and the proximate analysis.

It will be understood that the action of the chemical—phenol in this case—must be that of a true solvent and must not cause chemical changes either in its own structure or in that of the components of the coal. While there are several chemicals that will dissolve the solvent components of coal, phenol is best.

It does not seem that this method of coal analysis will be adopted generally in power plants, but our readers who are interested in that subject will find this bulletin well worth while.

STEAM CHARTS. By F. O. Ellenwood, Assistant Professor of Heat Power Engineering, Cornell University. Published by John Wiley & Sons, New York. Cloth, 7x9¾ in., 91 pages; 18 charts; 9 figures. Price, $1.

This book is intended to be of assistance to engineers and students when making calculations involving wet or superheated steam, and for that purpose the author has presented a set of charts convenient to handle and easy to read without extending the size of the charts beyond the dimensions of the page.

An introductory chapter sets forth the fundamental principles of pressure-volume and temperature-entropy diagrams, and another chapter is devoted to describing the preparation and use of the steam charts and a table of velocities, the scale of volumes being plotted from the values given by the steam tables of Marks and Davis.

A third introductory chapter defines atmospheric pressure and Baumétric corrections. There are an index chart, total, heat volume chart, external-work-volume chart, correction of mercury column for temperature and chart of correction of barometric readings due to change in elevation. There are also tables of correction of barometric readings and for capillarity, and tables of density of mercury and of theoretical velocities of steam expanding adiabatically in a frictionless nozzle.

Fifty illustrative problems are given with their solutions. These, together with the greater convenience of the charts over the large scale folders such as are usually employed for steam charts, render the task of making steam computation more inviting to the beginner and provide a work well adapted to the purposes of a handbook for engineers for data on the properties of steam and for checking methods of performing computations.

BUREAU OF STANDARDS PUBLICATIONS

Three instructive papers have recently been issued by the Bureau of Standards under title of "Measurement of Standards of Radiation in Absolute Value," "Various Modifications of Bismuth-Silver Thermopiles Having a Continuous Absorbing Surface," and "An Experimental Study of the Koepsel Permeameter," the last being an instrument for measuring the magnetic properties of iron and steel.

BOOKS RECEIVED

AMERICAN HANDBOOK FOR ELECTRICAL ENGINEERS. By Harold Pender. John Wiley & Sons, Inc., New York. Morocco leather; 2023 pages, 4¼x7 in.; fully illustrated; tables. Price, $5.

MACHINE SHOP PRACTICE. By Wm. J. Kaup. John Wiley & Sons, Inc., New York. Cloth; 199 pages, 5¼x7¾ in.; 158 illustrations. Price, $1.25.

HOW TO RUN AND INSTALL GASOLINE ENGINES. By C. Von Culin. Norman W. Henley Publishing Co., New York. Paper; 98 pages, 3¼x6 in.; illustrated. Price, 25 cents.

INSTALLING EFFICIENCY METHODS. By C. E. Knoeppel. The "Engineering Magazine," New York. Cloth; 258 pages, 7x10¼ in.; 103 illustrations. Price, $3.

TRADE CATALOGS

Burd High Compression Ring Co., Rockford, Ill. Directory of piston ring sizes. Illustrated, 68 pp., 4½x6½ in.

Elliott Co., 6910 Susquehanna St., Pittsburgh, Penn. Bulletin H. Alarm water columns. Illustrated, 8 pp., 7x10 in.

Harrison Safety Boiler Works, Philadelphia, Penn. Catalog No. 601. Cochrane multiport valves. Illustrated, 72 pp., 6x9 in.

Chicago Pneumatic Tool Co., Fisher Building, Chicago, Ill. Bulletin No. 34-K. Fuel oil and gas driven compressors. Illustrated, 24 pp., 6x9 in.

General Electric Co., Schenectady, N. Y. Bulletin No. 42,010. Small turbo-generator sets. Illustrated, 14 pp., 8x10½ in. Bulletin No. 42,300. Steam engine-driven generating sets. Illustrated, 12 pp., 8x10½ in. Bulletin No. 45,602. Lightning arresters for series lighting circuits. Illustrated, 8 pp., 8x10½ in.